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JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. 53. AUGUST 1941. No. 3

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SOME DEVELOPMENTS IN THE REPAIR BY WELDING OF FINISHED-MACHINED CASTINGS.

BY CARLTON G. LUTTS AND PAUL FFIELD.

Some "oldtimers" are going to find fault with this article. Eventually, as welding technique improves and the results are definitely proven in service, it is expected that repairs to finished-machined castings by welding will become a commonplace. In this article, the authors present actual cases where defects in steel castings, practically finished, have been repaired by welding with resultant saving of the casting and of most of the work expended on it.

A few years back every repair weld on a casting placed a stigma on that casting and left the impression that an inferior article was being substituted. This condition was still further aggravated because marine engineers were progressing towards higher

temperatures and pressures and consequently had to demand greater integrity from the component parts making up turbines and other high pressure steam units. Welding had not really come into its own and was usually considered by engineers to constitute a discontinuity in the piece welded very much as though the deposited material were merely a plug. Admittedly, many of the welds made in castings at that time were not a credit to the welding art and there are reputedly cases on record of welds " falling out of the casting."

Several changes were brought about which materially altered the welding of castings. Coated electrodes were developed, and specialized techniques of welding heavy castings were introduced. It was soon realized that to weld a heavy casting, while it was cold, irrespective of its analysis, was inviting trouble. Cracks and spalling were likely to occur even though the same steel in lighter sections was considered readily weldable. The importance of preheating and stress relieving and the judicious processing of the deposition of weld metal was becoming more thoroughly understood. The inevitable result of all these developments was the production of sound welds, superior to the casting which was being welded.

The developments in welding altered the light in which welding was regarded by marine engineers. It was realized that a sound structure consisting of a welded casting was preferable to an unwelded casting which might contain hidden defects. The Navy Department brought the question of sound cast structures to a point much nearer reality when they pioneered the use of radium for inspection of castings. The outstanding work of Briggs and Gezelius is too well known to require any further comment.

The Bureau of Engineering of the Navy Department assumed leadership with the foundries and shipbuilders in a concerted drive to produce better cast structures. Within liberal limits the Bureau of Engineering would permit the welding of many castings provided the repair was subsequently stress relieved and radiographed. The rejections of castings became fewer and fewer and without a doubt sounder castings were being obtained. Outright rejection of high pressure castings at Fore River has

amounted to less than four per cent in the last two years. It is believed that this experience is typical of other building yards.

The liberal attitude of the Bureau of Engineering towards welding soon produced such developments as cast structures especially designed for welding. At Fore River, were developed such castings as the high pressure turbine casing shown in Figure 1. This casting is built up of three separate castings welded into a single unit. The advantages for such a structure are numerous. However, the most desirable feature of this type of design is that it enables the foundryman to position each section in the most advantageous manner for making the casting. Sounder castings must result from such technique. It might be noted in passing, that the reduced price obtained from the foundry, because the casting was subdivided, slightly more than offset the cost of welding.

The welding of such structures as the built-up turbine casings afforded the Bethlehem Steel Company considerable welding experience and necessitated a certain amount of research concerning control and prediction of welding shrinkage. This experience seems to indicate that welding must cause distortion; that this distortion can be controlled and even offset, and finally that we have no cases on record of stress relieving causing distortion provided the work is reasonably supported and properly heated. This information has since formed the basis for still further developments in the welding of castings.

Even in the best regulated shops a defect will sometimes be revealed when the final machine cut is being taken on a casting. In the authors' opinion rarely are halfway measures applicable to such a condition. The defect must be excavated, repaired by welding, and the whole casting re-stress relieved. Halfway measures such as seal-welding the defect are hazardous. Furthermore, we now have strong indications that at the present operating temperatures the buried defect may creep through the seal weld, producing a leak. Repair by seal welding is then out of the question.

When such a condition arises the engineer must first determine whether the defect is serious. This can usually only be determined by radiography. It might be well to point out here that

the Navy's radiographic standards should not be considered as the sole means for judging the seriousness of such a defect. These standards were developed primarily as a general guide and some of the acceptable conditions "no repairs required" may not be acceptable if in a location of maximum temperature and pressure or at points of stress concentration.

Before citing several examples of finished castings which have been repaired, it might be well to clear up some of the controversial points concerning the hazards of such a process. Distortion of repair-welded castings is the result of movement caused by stresses which have been introduced during the course of the repair. The two sources of stress are shrinkage of the filler metal, and differential expansion arising from temperature gradients set up during the various phases of the welding process. Stresses originating from shrinkage of the weld metal are negligible in comparison to the stresses resulting from temperature differential. The magnitude and ultimate effect of thermal stresses are dependent principally upon such factors as preheating, quantity and rate of deposition of weld metal, location of the weld with respect to the casting, and method of subsequent stress relief.

Simple calculations show that the intensity of thermal stresses per degree of temperature difference is equal to (Modulus of Elasticity) $\times$ (Coefficient of Thermal Expansion). On this basis, for C-Mo steel the stress intensity per degree F. of temperature differential (at 400 degree) $= (30 \times 10^6)\ (7.9 \times 10^{-6}) = 237$ pounds p.s.i. Thus, with a temperature differential of 150 degrees F., a stress of 35,000 p.s.i. is raised, a value approaching the yield point of the material. Since the welding procedure involves the application of heat, it is necessary to consider each step in this welding in relation to establishment of temperature gradients.

The principal metallurgical reason for preheating a casting prior to welding is to prevent hardening and embrittlement of the weld area. But preheating serves not only to prevent embrittlement, but also to minimize the sharp temperature differential immediately established by deposition of liquid metal. The minimum preheating temperature depends upon the composition of the material and the amount of metal to be deposited. With C-Mo steel a minimum temperature of 400 degrees F. is used even when

small amounts of repair are to be made. When large amounts of metal are to be deposited, a higher preheating temperature is used in order to allow for heat lost by radiation.

During welding, the weld zone acquires a temperature far in excess of the casting itself, and thus the thermal expansion of this zone varies from points removed from the zone. On cooling, the weld zone cools at a faster rate and has a greater temperature range through which it must cool. All points do not cool through the same temperature range at the same time. It is through this variation in thermal expansion and contraction that internal stresses of high magnitude are set up. When they exceed the yield point, the material upsets upon itself, which leads to distortion if movement is unrestricted. The intensity of stress is especially high if variation in section exist in this area.

Location of the weld is also a factor in determining ultimate distortion. In a zone where movements become entirely localized, total distortion will be small. But if the zone is located in a position where small local movements become magnified into larger movements at a remote point by mechanical action, then the total distortion will be appreciable.

Although the process of stress establishment is inherent in the welding process, certain measures can be taken to minimize its magnitude and effect. Among these are control of welding rate, and judicious peening. Since distortion results directly from upsetting of the metal adjacent the welding zone, peening of each bead and the immediate surrounding area counteracts this upsetting. In addition, the weld area can be peened sufficiently to upset the metal in the opposite direction by an amount equivalent to the greater contraction which this area would normally undergo on cooling. Some care must be exercised in peening, however, since the casting has locked-up stresses which might easily lead to cracking. As will be explained later, peening may also be used in certain instances to correct distortion in repaired castings.

Another factor creating thermal stresses is quantity and rate of welding. Ideally, a minimum of distortion would result if the entire volume of weld metal could be deposited at once, or conversely, if the filler metal could be deposited in infinitesimally small amounts. Deposition of the entire quantity at once would

create an appreciable thermal stress, but the condition of stress would last for only a comparatively short time. By depositing a very small quantity of metal at a time, the temperature gradient is held to a minimum. It is by the use of intermediate welding rates that maximum distortion results, because a sizeable thermal stress is established and allowed to act for a period of time, thus causing movement and distortion in a manner similar to creep. In practice, the nearest approach to ideal conditions is obtained by employing a very small rate of feed. For example, less distortion is encountered by using a ⅛ inch wire, than with 5/32 inch or larger. By a combination of this lower welding rate and judicious peening, distortion can be held to a minimum.

Stresses locked up during welding are mostly dissipated during the stress-relieving operation. At a temperature of 1100 degrees-1200 degrees F. atomic readjustment and resultant negation of stresses occurs without distortion when certain precautions are observed. The principal precautions are proper support of work and maintenance of slow heating and cooling rates, with periodic soaks. For even in the stress-relieving operation new thermal stresses can result from non-uniform heating and cooling.

It is generally agreed that 400 degrees F. per hour divided by the maximum thickness in inches is a conservative heating rate, but there is an added factor of safety in employing one-half of this rate and this is recommended in important work. Unlike other heat treating operations the period of hold at temperature is not a linear function of thickness. Two hours at temperature is sufficient, for during the first hour the stresses of high intensity are reduced to those of the initially lower value, and after two hours all are reduced to a safe level. The work should not be removed from the furnace on the way down until a temperature of 400 degrees F. is reached, in order to avoid new thermal stresses.

The next point frequently raised is that oxidation or scaling is a factor to be considered. In a gas-fired furnace the losses of actual steel from scaling at the usual stress-relieving temperature have been negligible. Nothing other than a fine red powder has ever been observed. Even in electric furnaces operated without atmospheric control the amount of oxidation at 1200 degrees F.

is not excessive. Cleaning by kerosene or other measures removes the loose oxide and presents a surface equal to the surface of any turbine which has been run for some time. It is believed that the word " scaling " is responsible for most of the apprehensions from this source. Table I shows typical losses obtained from scaling in carbon-molybdenum and carbon steel castings at the upper limit of the stress relieving temperatures.

TABLE I.

LOSS IN THICKNESS OF STEEL DUE TO SCALING IN AN ELECTRIC FURNACE AT STRESS RELIEVING TEMPERATURES.

Time at 1250° F Hour	Average Reduction in Diameter	
	Mild Steel Inches	C-Mo. Steel Inches
1	0	.001
2	0	.001
4	.001	.001
8	.001	.001
16	.001	.001
24	.002	.002

Note: These losses are measured on the diameter of a 3/4 inch Rd. Bar. Losses on a plane surface would be one-half of the above amounts.

As a result of the fears of loss of metal from scaling, it has sometimes been proposed that on finished machined sections it is advisable to stress relieve at a lower temperature for a longer period of time. In the authors' opinion such a precaution is unnecessary and furthermore, is liable to lead to improper stress relieving. Stress relieving may almost be considered a short time creep treatment with certain fibers flowing plastically. Time and temperature are the conditions to be considered. What the relation between time and temperature is has not yet been determined as far as the authors know. 1200 degrees F. for one hour may be entirely adequate for a carbon-molybdenum steel. However,

at 900 degrees F., 5 hours or 5000 hours may not be sufficient. Until research has shown what the relationship is between time and temperature for stress relieving, the authors would not suggest lower temperatures and longer times.

In regard to actual repair of defective castings, the following paragraphs describe procedures used on three groups of castings. With the first group of castings, a certain amount of distortion could be tolerated, and no special precautions to avoid distortion were therefore required. The second group represents castings where re-machining could be used to correct a limited amount of distortion. The third group is representative of finished machined castings where no machining could be permitted after repair welding. Weld distortion with this group, therefore, had to be minimized and subsequently eliminated by means other than machining.

With respect to the first group of castings where a small amount of distortion can be tolerated, Figure 2 shows the high pressure steam inlet of a high pressure turbine where radiographic examination disclosed the defect shown in Figure 3. Since this location is in the region of maximum temperature, and since the stresses on the neck of the inlet pipe are usually rather indeterminate due to effect of connected steam lines, a complete excavation of the defective area and a sound weld were necessary. In this case minor distortion could easily be accommodated even if it meant refacing the inlet flange. Special precautions for this repair therefore, were not necessary. The casting was positioned for downhand welding and preheated over a generous area to 400 degrees F. The welding was deposited in and around the sides of the cavity, rather than uniformly filling the cavity. Each layer was peened partially to offset the welding shrinkage and to insure slag removal. Measurements were taken between the flanges of the inlet and the flanged joint of the casing before, during, and after welding. After welding, the casting was stress relieved at 1250 degrees F. It was not necessary to use an elaborate jig to support the casting. It was placed on a car bottom furnace and supported at various points with steel wedges as shown in Figure 4. After stress relieving, measurements showed that the casting had distorted less than .001 inch at any point. There was slight tarnishing of the machined surfaces and the paint on the casting

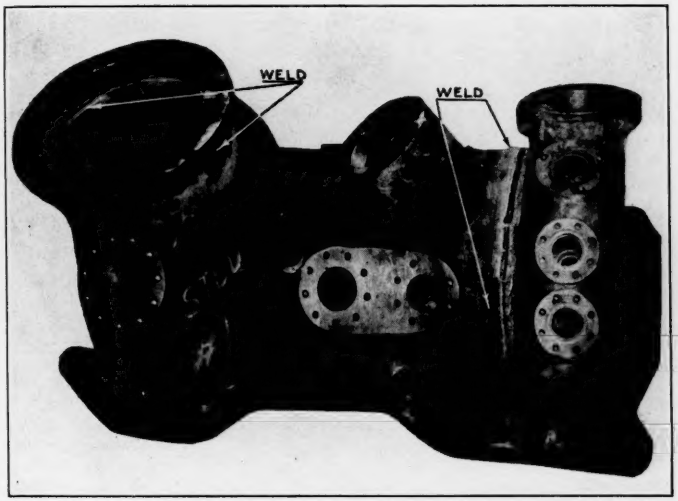


FIGURE 1.—H. P. TURBINE CASING COMPOSED OF THREE CASTINGS WELDED TOGETHER.

FIGURE 2.—STEAM INLET OF H.P. TURBINE CASING AFTER PARTIAL EXCAVATION OF DEFECTS SHOWN IN FIGURE 3.

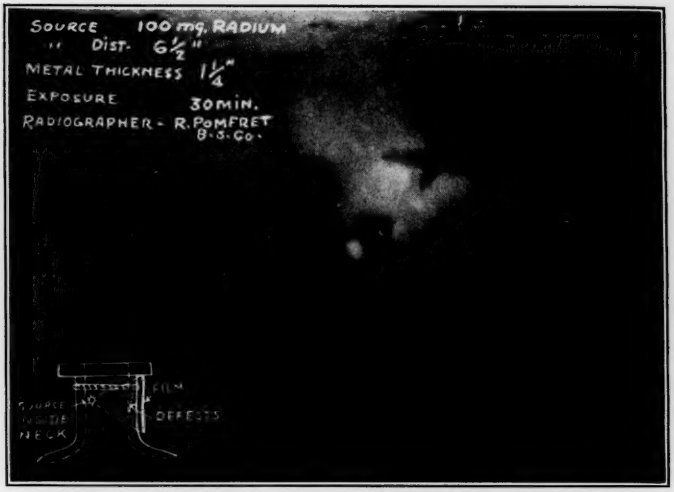


FIGURE 3.—RADIOGRAPH THROUGH DEFECTIVE AREA OF STEAM INLET SHOWN IN FIGURE 2.

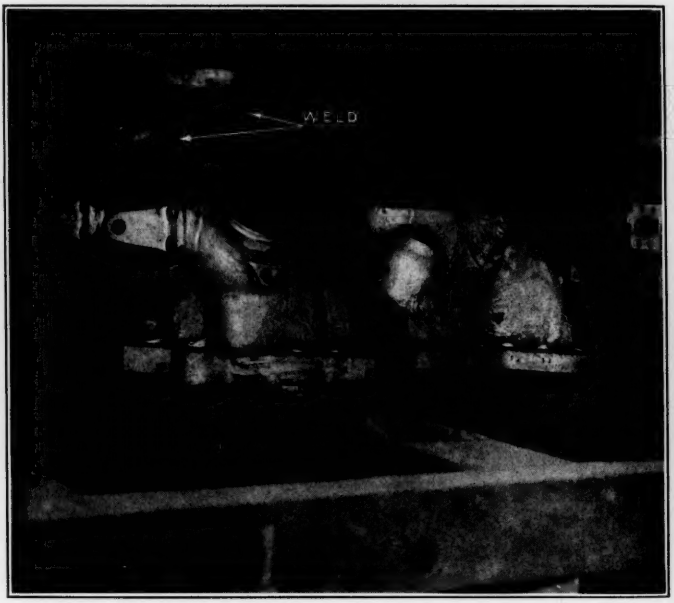


FIGURE 4.—REPAIR WELDED H.P. TURBINE CASING ON FURNACE CAR BOTTOM, AFTER STRESS RELIEVING.

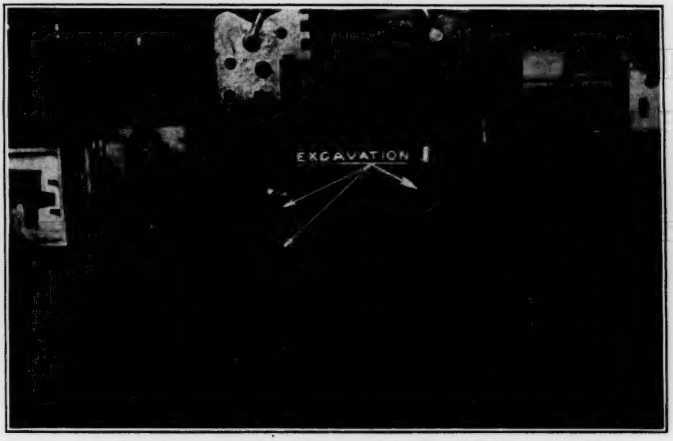


FIGURE 5.—ROUGH BORED H.P. TURBINE CASING AFTER EXCAVATION OF DEFECTS DISCLOSED DURING FINISH MACHINING OF FLANGES.

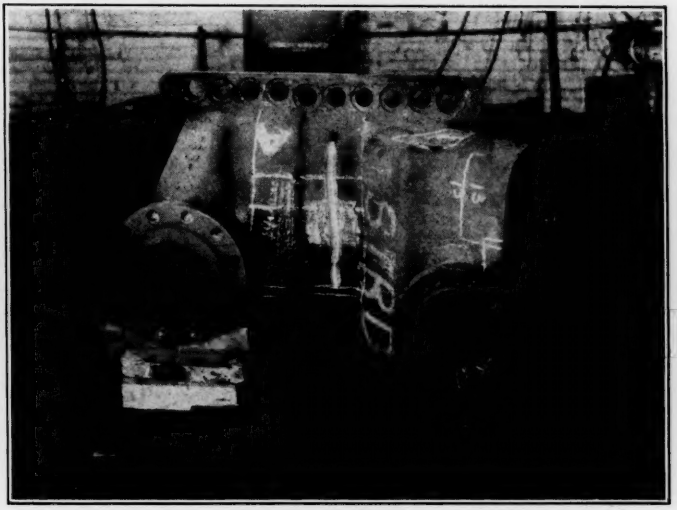
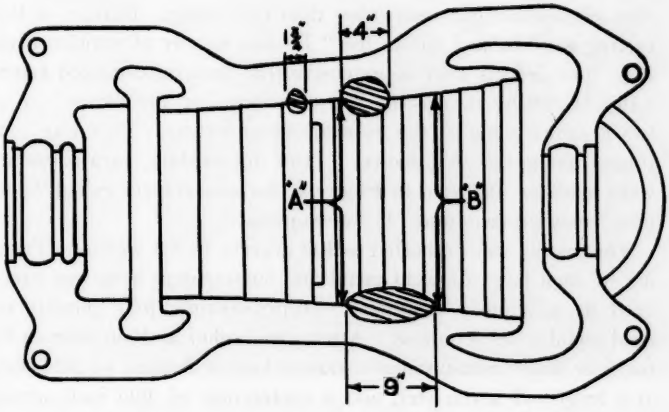


FIGURE 7.—PARTIAL EXCAVATION OF DEFECT IN H.P. TURBINE CASING.

was burned off, otherwise there had been no ill effects. The casting, however, was now definitely sound and no trouble from a hidden defect in this region would occur.

Figure 5 represents the second group of castings where distorted areas can be re-machined after repair. When the final cut was being taken on the flanges of this turbine casing, two cracks were disclosed which were not visible when the casting was rough machined. The casting had been rough but not finished bored. The bore was still 1/16 inch oversize, so warpage could be permitted which was within this limit. It would also be a simple matter to take a light cut off the flanges.

After cleaning out the cavities, weld metal was deposited approximately in the same manner as in the previous casting, and the usual precautions of preheating and cleaning were followed. The welding caused a convex deformation of the horizontal joint of approximately .007 inch. The bore of the casting also tended to widen, by approximately .012 inch. It can be noted by referring to Figure 6 that a slight recovery was obtained in the "B"



Before Welding "A"—21.986 inches "B"—23.003
After Welding "A"—21.997 " "B"—23.020
After Annealing "A"—21.999 " "B"—23.015
Welding caused a Concave Deformation of the Horizontal Joint, .007 inch.

FIGURE 6.—DISTORTION DUE TO WELDING OF H.P. TURBINE CASING.

reading in the bore after stress relieving this casting. After completion of the repair, the horizontal joint was re-machined to eliminate the convexity of .007 inch, and the casting was then finish-bored.

A second example of repair welding, where small amount of distortion can be corrected by re-machining is represented by an H. P. turbine casing shown in Figure 7. In this casting defects were present in the wall midway at the third staging, and also in the walls of steam chest. After removal of defects, welding was performed on the preheated casting. Each layer of weld-metal and adjacent area were peened and a total of thirty pounds of weld metal was deposited. The casting was stress relieved at 1200 degrees F. for two hours. Measurements after welding showed a high spot of .006 inch on one flange. A grinding operation restored required flatness to the casting.

An example of repair where no machining is permissible after welding is represented by an I. P. turbine casing shown in Figure 8. It can be noted that four rows of blading are in place. Distortion therefore had to be minimized during welding and then eliminated by means other than machining. Defects in this casting were located within the " T " slot in way of impulse blading. The defects were so extensive that removal consisted essentially of cutting the casting in two except for the flanges. Figure 8 shows some of the excavations underway. This was continued along the slot shown. Prior to welding, measurements were made as indicated in Figure 8, for convenience called " flatness " measurement and " C " measurement.

The casting was preheated to 400 degrees F. for welding. Peening of each layer of weld metal and surrounding area was practiced as previously described. Approximately fifty pounds of weld metal were deposited. After stress relief at 1050 degrees F. for four hours measurements showed loss of flatness of .014 inch in a length of seven feet, and a contraction of .005 inch across the " T " slot. It can be seen from the photograph that the contraction across the slot is the action which caused loss of flatness in the flange. In this casting, the width or diameter of the bore did not change beyond the small permitted tolerance.

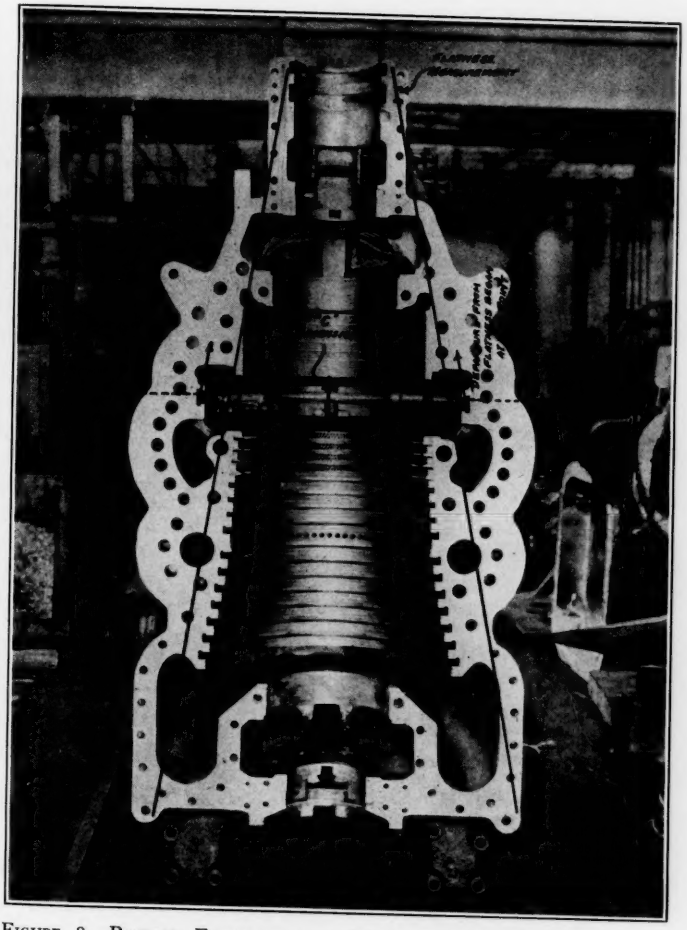


FIGURE 8.—PARTIAL EXCAVATION OF DEFECT IN I.P. TURBINE CASING.

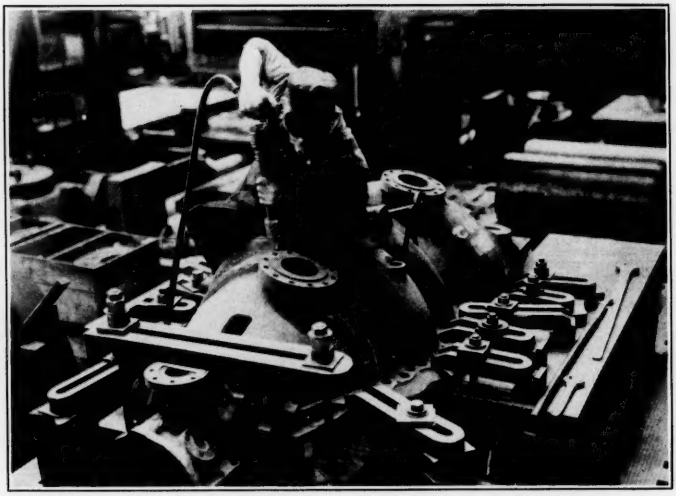


FIGURE 9.—METHOD OF RESTORING DISTORTED TURBINE CASING TO REQUIRED FLATNESS LIMITS.

Distortion was corrected in a manner shown by Figure 9. The casing was supported on blocks, and the small end of the casing was loaded so that the metal behind the " T " slot was put in tension. Peening of this area, as shown in the photograph, caused the metal to spread in a direction with the applied tension which was opposite to original weld shrinkage. Peening was continued until measurements showed that distortion was eliminated. The stress relief treatment given this casting did not cause any dimensional return, but in other cases it has been found necessary to straighten beyond the point finally desired, thereby allowing opportunity for slight dimensional return in stress relieving.

In summary, it has been shown that castings with large areas of finish machined surfaces whose flatness and dimensions are required to be maintained within close limits, can be successfully repair welded. Castings up to seven feet in length and weighing two tons have been repaired with as much as fifty pounds of weld metal added. Although problems connected with repair have differed somewhat with each particular casting observance of precautions previously described and use of approved welding technique have made it possible to repair successfully castings which in former years would have required rejection.

The authors wish to acknowledge the cooperation of Mr. J. E. Burkhardt, Technical Manager, Bethlehem Steel Co., Shipbuilding Division, whose advice and constructive criticism has been most helpful. Most of the development work on castings built up by welding has been due to his insistence on obtaining sound castings whether it be by design, by radiography, or by repair. The authors also wish to acknowledge the helpful cooperation of Rear-Admiral P. B. Dungan, formerly Inspector of Machinery at Fore River; Captain R. W. Paine, Engineering Superintendent, Boston Navy Yard; and Commander C. O. Kell, Production Manager, Boston Navy Yard, whose advice concerning repairs of steel castings has been most helpful.

ORGANIZATION AND FUNCTIONS OF THE BUREAU OF SHIPS.

COMMANDER D. H. CLARK, U. S. N., MEMBER.*

The Bureau of Ships will have entered its second year when this article appears. It is inevitable that organization changes will prove necessary as the Bureau matures, but there is every likelihood that such changes as may occur will be relatively minor. Since most readers of the Journal are interested, to a greater or less degree, in the Bureau of Ships or in its business, a description of the new Bureau is certainly in order. That following will give the reader a complete picture which can be improved only by personal observation.

The Bureau of Ships was established by Public No. 644—76th Congress, approved 20 June, 1940. This Act abolished the Bureaus of Engineering and Construction and Repair, and transferred the functions of these Bureaus to the Bureau of Ships. The object was to set up one organization, under one head, responsible for the design, construction and maintenance of naval vessels.

The remarkable thing about this consolidation is that it was not accomplished long ago. It appears worthwhile to examine the agencies in the Navy Department responsible for the design and construction of ships from the first steam propelled vessel in the United States Navy, until the present.

This first steam propelled vessel, which, by the way, was the first such vessel in any navy, was the *Fulton*. This vessel was designed complete—hull, engines and boilers,—by Robert Fulton. She was commenced in 1814 and completed in 1815. It is of interest to note that the hull was built at the shipyard of Adam

* Bureau of Ships.

and Noah Brown, on the East River, and was moved to Fulton's Machine Works on the North River, where the machinery was built and installed.

While the *Fulton* was being built, the Act of February 7, 1815, became effective, establishing the Board of Navy Commissioners under the Secretary. From then until 1842 the design and construction of vessels was carried on by naval constructors (civilians), under the direction of this Board.

Most of the vessels built during this period were sailing ships. In 1839, Congress authorized the construction of two steam frigates and two boards were appointed to determine the design and construction to be adopted,—one a board of line officers, and the other a board of naval constructors and an engineer. It is evident that the boards did not see eye to eye in the matter, for Secretary Paulding stated, "according to custom we have had boards sit and cogitate and disagree and compromise, so that in the end nobody will be responsible for a failure, if one should take place." Perhaps the Secretary was also the author of the remark that "Boards are long, narrow and wooden." *

The two steam frigates, the *Memphis* and *Missouri,* were completed in 1842, and marked the beginning of our steam navy. In that same year an Act was passed reorganizing the Navy Department, and establishing five bureaus. One of these, the Bureau of Construction, Equipment and Repairs, was assigned cognizance over design, construction and fitting out of naval vessels. This situation obtained from then until 1862, when the Department was again reorganized, and so for twenty years, the responsibility for ships was vested in one bureau. The Act of 1862 provided for eight bureaus, among which was the Bureau of Steam Engineering, and transferred the cognizance of the propulsion machinery to that Bureau. From then until June, 1940, the responsibility for the ships was divided among several Bureaus of the Department.

As the design and construction of naval vessels became more complex the need for closer coordination of the design and construction agencies was felt. In 1921, the Industrial Department of Navy Yards, which up to that time had consisted of a Hull

* Quoted from "History of the Construction Corps" by Rear Admiral W. P. Robert, (CC.) U.S.N., Retired.

Division and a Machinery Division, was reorganized to provide for a manager, a planning officer and a production officer, a move which facilitated the work of the Yards. In September, 1939, the Secretary of the Navy, recognizing the need for having one agency responsible for the design and construction of ships, appointed the present Chief of the Bureau of Ships, then the Chief of the Bureau of Engineering, as Coordinator of Shipbuilding. The consolidation of the two bureaus started then, and will continue for some time. Much progress has been made to date, and there is no question but that the solution of that most complex engineering problem, the design and construction of the modern naval vessel, has been facilitated by the merger.

Figure 1 is an organization chart of the Bureau. This will assist in following the organization and functions of this rather large, and at present, still rapidly growing organization. (As of 1 February, it consisted of 182 officers and 1318 civilians, exclusive of field activities.) Full lines on this chart indicate channels of direct administrative control and technical authority. Broken lines indicate technical authority, but without direct administrative control.

The Bureau consists of three independent offices, five divisions and a number of technical sections. The technical sections, which are under the administrative control of the Design Division, are responsible to all the Divisions of the Bureau for matters under their cognizance. For instance, the Boiler Section will recommend the design of boilers for a naval vessel to the Design Division; will pass on the design of boilers for a merchant vessel to the War Plans Division; will recommend the award of a contract for boilers for a navy yard built ship to the Shipbuilding Division; and will recommend concerning alterations or repairs of boilers of naval vessels in commission and finally accepted, to the Maintenance Division.

The names of the divisions and subdivisions of the Bureau indicate broadly the functions for which they are responsible. However, some elaboration is necessary to an understanding of "how the wheels go around."

The General Inspector supervises the administration of the offices of the Supervisors of Shipbuilding, Inspectors of Ma-

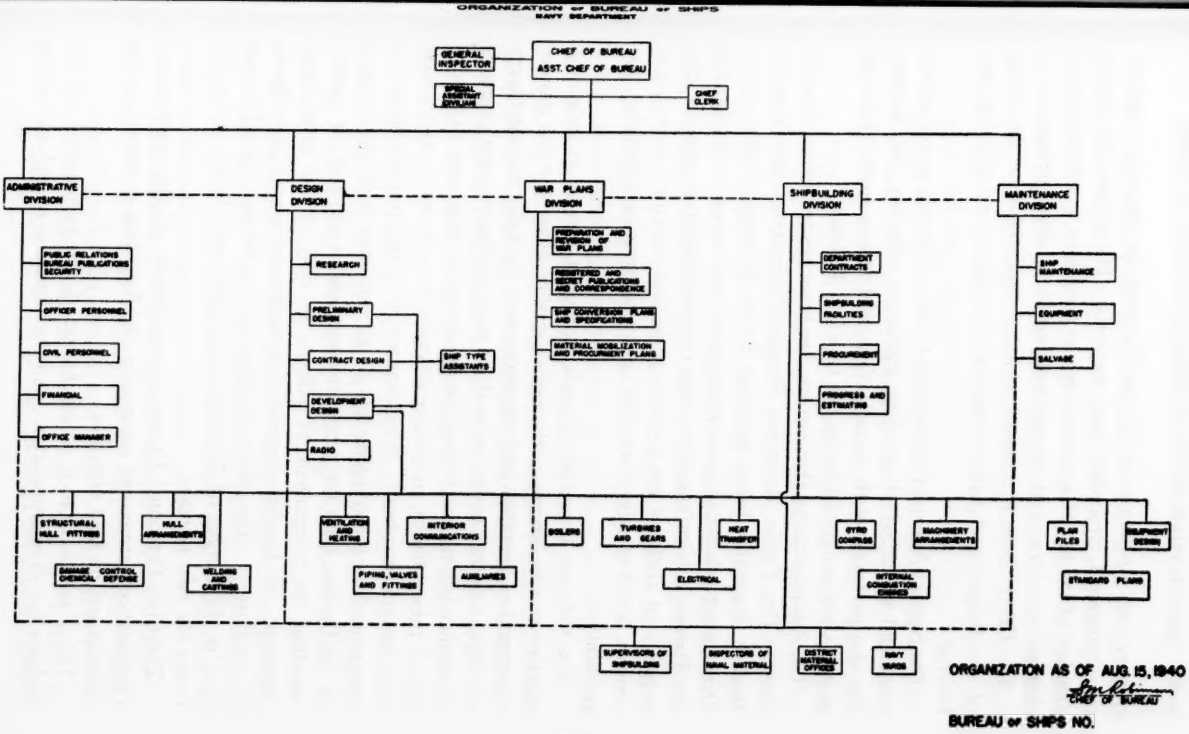

ORGANIZATION OF BUREAU OF SHIPS
NAVY DEPARTMENT
GENERAL INSPECTOR
CHIEF OF BUREAU
ASST. CHIEF OF BUREAU
SPECIAL ASSISTANT (STYLING)
CHIEF CLERK
ADMINISTRATIVE DIVISION
DESIGN DIVISION
WAR PLANS DIVISION
SHIPBUILDING DIVISION
MAINTENANCE DIVISION
PUBLIC RELATIONS BUREAU PUBLICATIONS SECURITY
OFFICER PERSONNEL
CIVIL PERSONNEL
FINANCIAL
OFFICE MANAGER
RESEARCH
PRELIMINARY DESIGN
CONTRACT DESIGN
DEVELOPMENT DESIGN
RADIO
SHIP TYPE ASSISTANTS
PREPARATION AND REVISION OF WAR PLANS
REGISTERED AND SECRET PUBLICATIONS AND CORRESPONDENCE
SHIP CONVERSION PLANS AND SPECIFICATIONS
MATERIAL MOBILIZATION AND PROCUREMENT PLANS
DEPARTMENT CONTRACTS
SHIPBUILDING FACILITIES
PROCUREMENT
PROGRESS AND ESTIMATING
SHIP MAINTENANCE
EQUIPMENT
SALVAGE
STRUCTURAL HULL FITTINGS
DAMAGE CONTROL CHEMICAL DEFENSE
HULL ARRANGEMENTS
WELDING AND CASTINGS
VENTILATION AND HEATING
PIPING, VALVES AND FITTINGS
INTERIOR COMMUNICATIONS
AUXILIARIES
BOILERS
TURBINES AND GEARS
ELECTRICAL
HEAT TRANSFER
GYRO COMPASS
INTERNAL COMBUSTION ENGINES
MACHINERY ARRANGEMENTS
PLAN FILES
STANDARD PLANS
EQUIPMENT DESIGN
SUPERVISORS OF SHIPBUILDING
INSPECTORS OF NAVAL MATERIAL
DISTRICT MATERIAL OFFICES
NAVY YARDS
ORGANIZATION AS OF AUG. 15, 1940
CHIEF OF BUREAU
BUREAU OF SHIPS NO.

FIGURE 1.

chinery and Inspectors of Naval Material, as regards organization, personnel, procedure and methods. He is also the senior member of the board appointed by the Secretary to adjudicate the changes in contracts of vessels building by private contractors.

The Chief Clerk is charged with the supervision of the clerks of the Bureau and with reviewing requisitions for printing and binding.

The Special Assistant advises the Chief of Bureau as to the contents and intent of all pending legislation affecting the Bureau. He also performs such investigations and makes such special reports as are assigned him by the Chief of Bureau.

The Administrative Division has five branches as indicated on the chart. The Public Relations, Bureau Publications and Security Branch, maintains liason with the Public Relations Office of the Department relative to press releases and interviews; is responsible for Bureau publications other than technical specifications; handles requests of civilians for visits to ships and stations; reviews and censors articles, advertisements and photographs submitted for publication.

The Officers Personnel Branch maintains files of information relative to the officers with special qualifications for the design, construction, operation and maintenance of the hulls and machinery of ships, whether active, retired or reserve officers; and prepares recommendations concerning the appointment education and training of these officers, or on any disciplinary matters in which they may become involved. It is charged with the responsibility for recommending the complements of all activities under the control of the Bureau, both for peace and war conditions; and for recommending the assignment of officers to duty to fill these complements. It handles requests for temporary travel for officers of the Bureau. The officer in charge of this Branch has additional duty in the Bureau of Navigation, and performs liason duties between the two Bureaus.

The Civil Personnel Branch is responsible for the recruitment, classification rating, and promotion of all civilian personnel in the Bureau and its field activities.

The Finance Branch is in charge of the preparation of the budgets for all appropriations, and the allocation of and accounting

for the funds appropriated. If you have not built up an immunity to numbers following the dollar sign, the fact that this office is handling approximately $1,250,000,000 or 28 per cent of the total Navy appropriation for the fiscal year 1941, and about $1,500,000,000 or, 42 per cent of the expected total Navy appropriation for the fiscal year 1942, may impress you with the extent of the activities of this office and of the Bureau.

The Office Manager is responsible for the organization and administration of the service groups of the Bureau, i.e., the Mail Section, Filing Section, Correspondence Section and Messenger service. He is responsible that all correspondence and communications are properly routed, and move through and out of the Bureau in the minimum of time.

The War Plans Division is made up of four branches as indicated on the chart. The first branch is responsible for the preparation, revision and distribution of the Material Contributory Plans for the Bureau of Ships, and for reviewing, correcting and approving the contributory plans of the Naval Districts.

The next branch of this Division is custodian of all registered and secret publications and correspondence in the Bureau.

The Ship Conversion Plans and Specifications Branch develops plans for the conversion of merchant ships into naval auxiliaries, and reviews the designs of all merchant vessels proposed by the Maritime Commission to insure maximum usefulness for National Defense.

I believe it wise to say that the functions of the Material Mobilization and Procurement Plans Branch are indicated by its name.

The Shipbuilding Division is made up of four branches, as follows:

The Department Contracts Branch, which handles all legal matters involving the Bureau, comments on existing and proposed legislation involving the Bureau and recommends concerning all matters relating to Department Contracts.

The Shipbuilding Facilities Branch maintains records of existing and projected Class I shipbuilding facilities in the United States; studies and makes recommendations in the cases of private companies requesting financial subsidization; handles the Bureau's work in connection with the Shore Station Development Board,

and the Annual Machine Tool and Equipment projects; recommends the allocation of major shipbuilding projects to private and Government Yards.

The Procurement Branch handles all matters relative to requisition, award, inspection, delivery, stocking, disposition and shipping of materials and equipment, excepting certain special materials.

The Progress and Estimating Branch reports the progress of ships under construction or conversion; schedules the priority of materials in connection therewith; maintains an analysis of delays and reports prospective delays; maintains cost records and prepares estimates on projected shipbuilding; and prepares monthly reports of progress for vessels building. There are in the Branch six type desk assistants for the various classes of ships who are responsible for handling all matters with which the Branch is charged, for the type or types assigned. These officers direct all matters, other than design, in connection with the acquisition and conversion of vessels for the navy.

The Design Division, which is the largest of the five divisions, and represents more than half of the officers and civilians assigned to the Bureau, is made up of five branches, the ship type assistants and the technical sections.

The Research Branch is composed of a Research Section and a Standards and Tests Section. The Research Section administers research projects initiated by the Bureau; reviews the reports of these and other reports, technical papers, articles, and ONI reports of interest to the Bureau, and makes them available to the various interested Branches and Desks. It maintins the Bureau's Technical Library. The Standards and Tests Section is responsible for the administration of the various test laboratories and for the preparation and distribution of the General Specifications, Navy Department Specifications, Supplementary General Specifications and ad interim leaflet specifications for materials.

The Radio and Sound Branch is responsible for research, development, manufacture, test, installation, and maintenance of radio and underwater sound code and cipher, and other than orthodox electrical signaling equipment for ships, shore stations and aircraft.

The other three branches of the Design Division are more directly related to the design of ships. The early stages of the design are handled by the Preliminary Design Branch. This Branch assists the General Board in arriving at an optimum set of characteristics and prepares various studies as requested. The General Board calls hearings on the new design which are attended by the representatives of the various Bureaus and offices of the Department. When all phases of the design have been discussed and studied, the General Board draws up a directive outlining the military characteristics required in the ship. The Preliminary Design Branch then establishes the lines and body plan, and develops general arrangement plans covering major arrangement features, makes strength and protection calculations to permit a weight estimate. The model of the lines is towed to give a closer estimate of shaft horsepower for the desired speed and to determine if revision of the lines is necessary. This Branch checks to determine that stresses are acceptable, that lines are satisfactory, prepares stability curves for the various conditions of loading, floodable length calculations, and investigation of transverse stability in damaged condition. When all these have been checked, and it is reasonably certain that the ship can be built to the required characteristics, the design is turned over to the Contract Design Branch. This Branch is made up of the Hull, Machinery, and Electrical Drafting Rooms; Ship Specification Sections, Hull and Machinery; Weight Section, Stability and Computing Section (Hull), Head Balance Section, Applied Mechanics Section, Propeller Section and the several officers to coordinate and administer these.

The Contract Plans and Specifications for the ship are prepared in the Contract Design Branch. In their development, comments are requested from the various Bureaus on the design and approval obtained from the Secretary as to compliance with the General Boards directive. The contract plans include:

Lines and Offsets
Outboard and Inboard Profiles
Midships and typical sections
Plating, outside and decks

Framing
Arrangement Plans (decks)
Ballistic Plating and Connections
Turrets
Ammunition Stowage and Handling
Piping diagrams
Ventilation diagrams
Machinery Arrangement Plans
Electrical Plans
Rudder and Steering Gear.

The preparation of these plans requires from three to four months for a large ship and involves many calculations to determine weights, strength, centers of gravity, etc. In developing these plans and specifications, the Branch makes full use of the technical sections and the ship type assistants which you will note on the chart are available to Preliminary, Contract and Development Design Branches.

When completed, bids are invited, or a contract is negotiated, by the Secretary through the Shipbuilding Division, and the Judge Advocate General's Office and the construction period commences. From then until the completion of the vessel, and its acceptance after final trials, it comes under the Development Design Branch. The working plans as developed are reviewed by the Supervisors of Shipbuilding. Many are acted on by these offices and others are forwarded to the Bureau, where they are acted on by the Ship Type assistants and the various technical sections. The technical sections insure that the best designs are obtained in their own specialties, and the ships' type desk officers insure that these are coordinated into the best possible whole ship. Some changes in the specifications prepared by Contract Design are always resorted to during this building period. These are handled as changes in, or developments under the contract, and if changes in cost are involved, they are adjudicated by the Board of Officers appointed by the Secretary, of which the General Inspector is the senior member. When the ship is completed, preliminary trials are held under the direction of the Board of Inspection and Survey, and work items based on the recommendations of the Board are undertaken. The ship then takes a shakedown cruise, after which final

trials are held, and other work items recommended by the Board are accomplished. Some of these items are the responsibility of the contractor and must be accomplished to fulfill the contract. Others are Government responsibility and are undertaken to improve the military value of the ship. These latter items are not a charge against the contractor. Following satisfactory completion of the final trials, the vessel is accepted, and is then transferred to the cognizance of the Maintenance Division, where it remains until stricken from the Navy list.

The Maintenance Division is divided into a Ship Maintenance Branch, an Equipment Branch, and a Salvage Branch. This Division is responsible for all matters relating to the operation and maintenance of the naval vessels assigned to it, including repairs and alterations. It handles ships' allowances, surveys of material, stock and issue of equipment, including the special hull and machinery materials not handled by the Shipbuilding Division or the Radio Branch. It handles ship salvage matters.

The Ship Maintenance Branch is composed of Ship Type Desks, similar to those of Design Divisions. The officers assigned to these desks are responsible for all vessels of the type under the Maintenance Division. They are made to feel that they are the personal representatives in the Bureau of the ships assigned their desks. In discharging their duties, they consult freely with the technical sections and other divisions and branches. The following list contains some of the items for which they are responsible for taking action:

1. Arrival conference and departure reports of Navy Yard overhauls.
2. Requests for repairs and alterations.
3. Allotment of maintenance funds, in some cases.
4. Application to the Equipment Branch for movement of special materials.
5. Requests for special materials.
6. Preparation of the desk budget.
7. Maintaining of up-to-date records of alterations showing those completed and those outstanding and budgetary requirements.

8. Review of Material Inspection Reports and coordination of Material Improvement program.
9. Recommends to the Equipment Branch on all ships' allowances.

If at this point you do not feel that you could take over the direction of a Division or Branch of the Bureau, do not be harsh with either yourselves or me. It is a sizable organization, doing many and varied things, some of which are quite complex.

EXPERIMENTAL HIGHLIGHTS OF VISUAL EXPERIMENTATION WITH CENTRIFUGAL PUMPS.

LT. COMMANDER HOWARD L. COOPER, U.S.N.R.

Our present national emergency accentuates the necessity of utilizing to the greatest practical extent experimental developments in all fields. This same impetus should encourage further experimental investigation, ever advancing the perfection of our works and enriching our knowledge. Only through diligent collective cooperation may we bring our national program to a successful conclusion. As a reward, we hope to achieve the perpetuation of our manner of living, enhanced by an elevated standard of living, or more and better goods for more people, with the satisfaction of a job well done.

The ensuing paper presents a descriptive discussion of some of the experimental work carried out by the author while with the U.S.E.D. during the years 1936 and 1937. It concerns a fact-finding model study in connection with a 28-inch dredge pump. This type of pump has given excellent service on various government projects.

The discussion is developed for the purpose of acquainting the reader with the high lights of visual experimental work and points out possible applications in other fields. For the sake of brevity and compactness, it will be necessary to condense the description of the more common test equipment. This is facilitated by the use of photographs demonstrating general layouts.

A general list of the topics covered is included below:

1. Description of Models and Layout.
2. Velocity and Pressure Distribution.
3. Observation of Flow Conditions.
   (a) Action of Rubber Balls.

4. Factors Influencing Efficiency.
5. The Experiment with Tongue Extension.
6. Similitude and the Acting Forces.
7. General Conclusions.

Description of Models and Layout.

A model scale of 1 : 6 was selected as the most satisfactory solution to the combined requirements of transparency, economy, simplicity of construction, dynamometer-power limitation and similitude requirements. The volute was constructed of heavy pyralin, mounted in a rigid cast aluminum frame. The impellers consisted of heavy pyralin shrouds securely mounted on cast aluminum spiders, which in turn were keyed on the pump shaft. Pyralin is a plastic material composed of cellulose nitrate, possessing excellent transparency and fulfilling the physical requirements.

The volute described above was built in accordance with the latest design of a 28-inch Army Engineer's Dredge Pump and for identification purposes is designated as Pump Casing " A ". For similar reasons, the designation Impeller " A " was given to the model impeller corresponding to the latest design used in this Pump. Both pump casing " A " and Impeller " A " are shown in Figure 1.

Impeller " B " was adapted to the 1 : 6 Dredge Pump Model from an earlier design for the 1 : 4 Hopper Dredge Pump Model. The outstanding difference between the two impellers is the difference in the exit angles. In the case of Impeller " A ", the exit angle of the fluid is 49° 40′ while that of Impeller " B " is 63° 30′. In both cases the entrance angles of the impeller blades are 42° 30′.

The exit angle of Impeller " C " is identical with that of Impeller " B ". However, this impeller was designed to give a much more gradual progression of area between the suction nipple and periphery of the Impeller.

Figure 2 shows the complete testing setup as used in obtaining the experimental data. No valves existed in either the suction or discharge lines. The model was operated with the water surface below the centerline of the pump in conformity with the prototype. Nozzles (or orifices), were used at the end of the discharge line to simulate a wide range of pipeline lengths. Discharge measure-

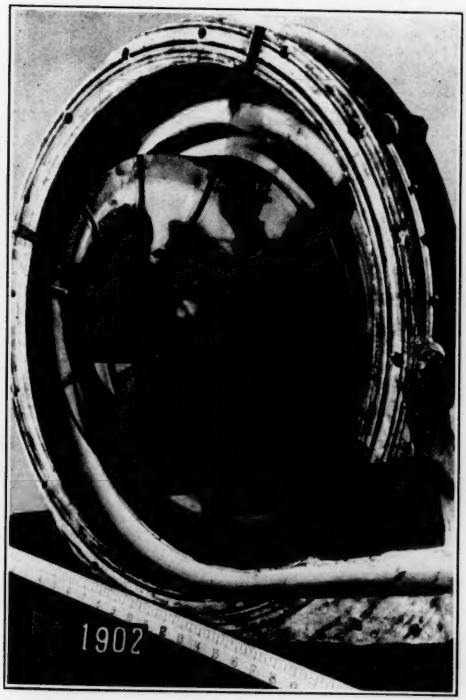


FIGURE 1.—BACK HEAD OF PUMP CASING "A," IMPELLER "A."

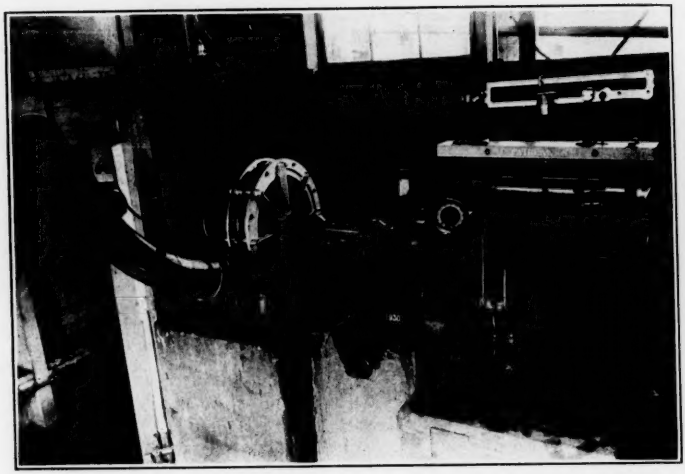


FIGURE 2.—ASSEMBLED PUMP, PIPING LAYOUT 11, AND DYNAMOMETER AS USED FOR EXPERIMENTS.

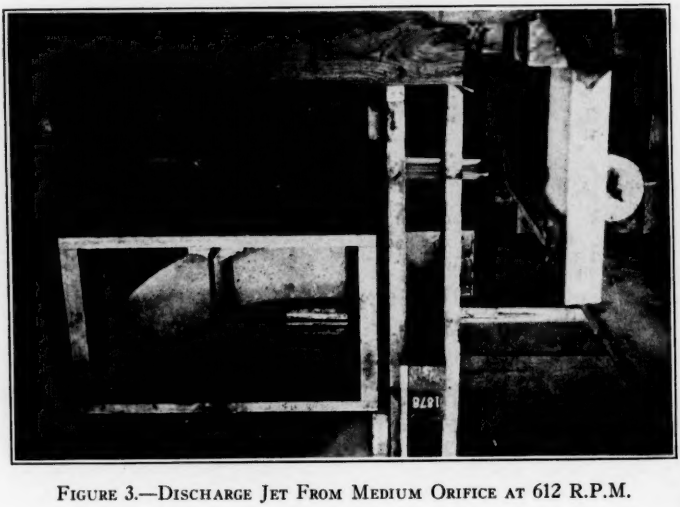


FIGURE 3.—DISCHARGE JET FROM MEDIUM ORIFICE AT 612 R.P.M.

ments were made either with a weir or volumetric tank in addition to nozzles or orifices and checked against area times velocity results.

The problem of simulating definite prototype pipe line lengths was of considerable interest. In order to do this, it was necessary to assume one set of operating conditions.

The necessary assumptions made for prototype conditions were:

Pump: 28-inch Dredge Pump with a 78-inch diameter Impeller and a 31-inch suction eye.

Fluid: Clear water, containing no solids.

Speed: 250 R.P.M. This corresponds to a peripheral speed of

$$3.14 \times \frac{78''}{12} \times 250 \text{ R.P.M.} \quad 5100 \text{ F.P.M. or } 85 \text{ F.P.S.}$$

Suction line: 33 inches diameter.

Discharge line: 28 inches diameter, various lengths.

Suction Head: 14 inches of mercury or 15.9 feet of water at 80 C.F.S. of which 6.0 feet are static lift.

Total Static Lift: 15.0 feet at 250 R.P.M.

Coefficient " n " = 0.012 in the Manning formula for pipe losses.

$$h_{fd} = \frac{4.66\, n^2 L Q^2}{d^{16/3}} \text{ (Friction head in feet of water)}$$

L = Pipe line length in feet

d = Internal pipe diameter in feet

Q = Discharge in C.F.S.

The value of the coefficient "n" was found to be about 0.0095 on an important government project. This pipe line is exceptional, being stationary and of most careful construction. Pipe lines representing average field conditions are often of floating construction and irregular nature, indicating a rather conservative value of " n ".

The substitution of the assumed values for "n" and "d" in Manning's formula gives the following equation for the discharge line friction losses:

$h_{fd} = 0.000007311\ LQ^2$ ..... (1)

L = Length in feet of 28-inch pipe line.

Q = Discharge in C.F.S.

Using the remaining prototype assumptions, the equation for total head H in feet developed by the pumps for a given discharge becomes:

$$H = h_{fd} + 6 + 9.9\left(\frac{Q}{80}\right)^2 \text{ (Suction head)}$$

$$+ 9 \quad \text{(Discharge line lift)}$$

$$+ \frac{V_d^2 - V_s^2}{2g} \quad \text{(Velocity head correction)} \text{ ..... (2)}$$

Additional assumptions necessary for the model are:

Scale of the model equals 1:6; applicable to all linear dimensions including static lift.

$$\text{Ratio of head} = \frac{\text{model head}}{\text{prototype head}} = 1:6$$

Ratio of Speeds: Since the static head developed by a centrifugal pump varies with the square of the peripheral velocity, the latter must be reduced in the model in the ratio of $1:\sqrt{6}$, or to $\frac{85 \text{ F.P.S.}}{2.45} = 34.7$ F.P.S. On the other hand since the diameter of the impeller is reduced in the model in the ratio of 1:6, it is therefore $\frac{78 \text{ inches}}{6} = 13$ inches or 1.083 feet, and its speed must be

$$\frac{34.7 \text{ F.P.S.}}{3.14 \times 1.083 \text{ feet}} \times 60 \text{ secs.} = 612 \text{ R.P.M.}$$

Ratio of Discharges: Since the areas vary with the square of the linear dimensions, in the ratio $6 \times 6 = 36$ and the velocities vary as $\sqrt{6}$, the discharge ratio must $= \frac{1}{36\sqrt{6}} = \frac{1}{88.18}$

Model Suction Line: To be constructed so as to cause 1/6 of the prototype suction head.

Model Discharge Line: To be so designed as to make the sum of the discharge line friction losses plus the head loss at the orifice equal 1/6 of the friction loss in the prototype discharge line.

Head Loss at Orifice: To be equal to the deflection at the orifice as computed by the Byron Jackson formula.

$$G.P.M. = 20\, D^2 \text{ in. } \sqrt{h_{ft}} \propto M$$

G.P.M. = discharge in gallons per minute.

D = diameter of discharge line in inches

h = head or deflection in feet.

$$\propto = 0.598 + 0.422\, M^2$$

$$M = \frac{d^2}{D} = \frac{(\text{orifice diameter})^2}{(\text{pipe diameter})}$$

From equation (1), the orifices may be designed by assuming a given discharge for each pipe line length.

After determining the discharge and the discharge line friction loss for each pipe line length, these data and all other prototype assumptions must be reduced to model scale values. The model equivalent of the prototype discharge line friction loss is the sum of the actual model discharge line losses plus the head loss at the orifice. The actual losses in the model discharge line were estimated on the basis of the following assumptions:

1. All losses vary with Q^2.

2. Pipe friction losses computed by Manning formula with $n = 0.012$.

3. Loss in expansion from 4.67 inches to 7.02 inches $= \dfrac{0.25v^2}{2g}$

4. Loss in two 45 degree bends $= \dfrac{0.5v^2}{2g}$

From the tabulation of model equivalent values for the various pipe line lengths simulated, the necessary orifice deflection was computed for each desired condition. When model values of G.P.M., D, and $\sqrt{h}$ are substituted in the Byron Jackson equation, it assumes the form of $AM + BM^3 = C$. This equation may now

be solved graphically for each desired prototype pipe line condition with sufficient accuracy to determine the orifice diameter ($d = D\sqrt{M}$) to the nearest 0.001 inch.

After the completion of the efficiency tests, the model data were analyzed to determine the magnitude of the error involved in the model loss assumptions. The results verified the assumption that they varied with Q^2. However, the actual value of the losses were found to be somewhat less than assumed. Table 1 represents the final results of the orifice design computations as adjusted to actual losses in the model. It should be noted that the results shown are for total equivalent length and include no bends and that they hold only for assumed prototype conditions.

TABLE 1

ORIFICES USED TO SIMULATE VARIOUS LENGTHS OF PROTOTYPE PIPE LINE

Orifice:	Diameter in inches	Length of Prototype 28 inch Pipe Line Simulated	
		n=0.012	n=0.0095
0	0.000	infinite feet	infinite
1	1.756	15,525	24,840
2	1.963	9,925	15,880
3	2.145	6,925	11,080
4	2.332	4,925	7,880
5	2.464	3,925	6,280
6	2.646	2,925	4,680
7	2.926	1,925	3,080
8	3.141	1,425	2,280
9	3.471	925	1,480
10	3.787	625	1,000
11	4.194	385	616
12	4.660	225	360
13	5.593	75	120

Discharge measurements were made by means of the discharge jet deflection method and also the volumetric tank method.

Orifices were so designed and installed as to comply with the specifications of the International Standards Association for orifice measurement of discharge. Orifice deflections up to 10 feet were measured on a water gage to an accuracy of $\pm$ 0.03 feet. Deflections over 10 feet were measured by a specially designed and calibrated mercury gage to an accuracy of $\pm$ 0.05 inches. As the discharge varies with the square root of the deflection, the error due to reading the deflection cannot exceed 0.3 per cent.

Figure 3 shows the discharge jet and deflection for a medium orifice opening.

For primary discharge measurements, the volumetric tank method was used in preference to weir measurement for the purpose of checking orifice discharge measurements on account of its greater accuracy.

A directly coupled 15 H.P. Diehl Dynamometer was used to drive the model. The dynamometer scale measured the armature reaction to the nearest 0.1 pounds. By construction, the lever arm from the centerline of the motor to the knife edge was fixed at 12.6 inches. With a reaction of W pounds and a speed of

$$\text{N in R.P.M., the brake H.P.} = 2\pi \frac{12.6}{12} \frac{WN}{33000} = \frac{WN}{5000}$$

A reasonably constant voltage supply was furnished by a separately excited D.C. generator with a sensitive means of control over a wide range of speeds. A recording tachometer indicated approximate speed and total revolutions in any given period of time.

The measurement of brake horsepower introduced the largest single source of error in the test data as it was impossible to avoid charging clutch, bearing and packing gland losses to the pump. The percentage loss sufficiently accentuated at low speeds to render worthless prototype brake horsepower and efficiency predictions. At high speeds these losses were reduced to approximately 1 per cent.

Velocity measurements were made with a modified Iowa Type Pitot tube with a streamlined velocity orifice. In order to reduce the effect of oblique currents from the impeller on pressure readings, the usual single pressure orifice at the bottom was replaced with two pressure orifices at the sides. The velocity was

read directly from differential gages calibrated by the formula, $v = 6.4\ H^{.526}$, which was determined at the Institute of Hydraulic Research at Iowa City.

All pressure measurements were determined with the aid of manometers, using water manometers for low pressures and mercury manometers for high pressures.

Velocity and Pressure Distribution.

Eight sections were selected for velocity observations, one in the suction nipple, another in the discharge nipple, and the remaining six in the volute. The location of the sections was governed by the construction features of the model and the anticipated relative importance of the observations. Two or more openings were provided in each section for inserting the Pitot tube; velocity readings were taken at ¼-inch intervals in each of the 22 traverses for which openings had been provided.

Using Pump Casing "A" and Impeller "A", velocity distributions were obtained from observations at all points and plotted as velocity contours illustrated in Figure 4. A study of this exhibit indicates several points of particular interest. In most sections the general effect of increasing the discharge is to lower the velocity gradient along radial traverses. This is especially noticeable in section GG where the average velocity in the bottom of the volute increases over 50 per cent while the velocities near the impeller decrease slightly. The flow in the discharge nipple is poorly distributed for small orifices (long pipe lines), and in the case of orifice 2 (9925 feet), was actually reversed in the bottom of the nipple.

A comparison of the observed velocities in the suction and discharge nipple and those in the volute brought forth one of the most important discoveries obtained in the course of the experimental work. The comparatively low velocities in the suction nipple are increased about 6 times in the volute only to be slowed down greatly in the discharge nipple. After computing the quantities of flow through each section, it is concluded that considerable recirculation of fluid occurs within the pump. A graphical illustration of this phenomenon for each section is given in Figure 4.

In pumps of this nature where the fluid is loaded with a high percentage of abrasive material, both recirculation and high volute

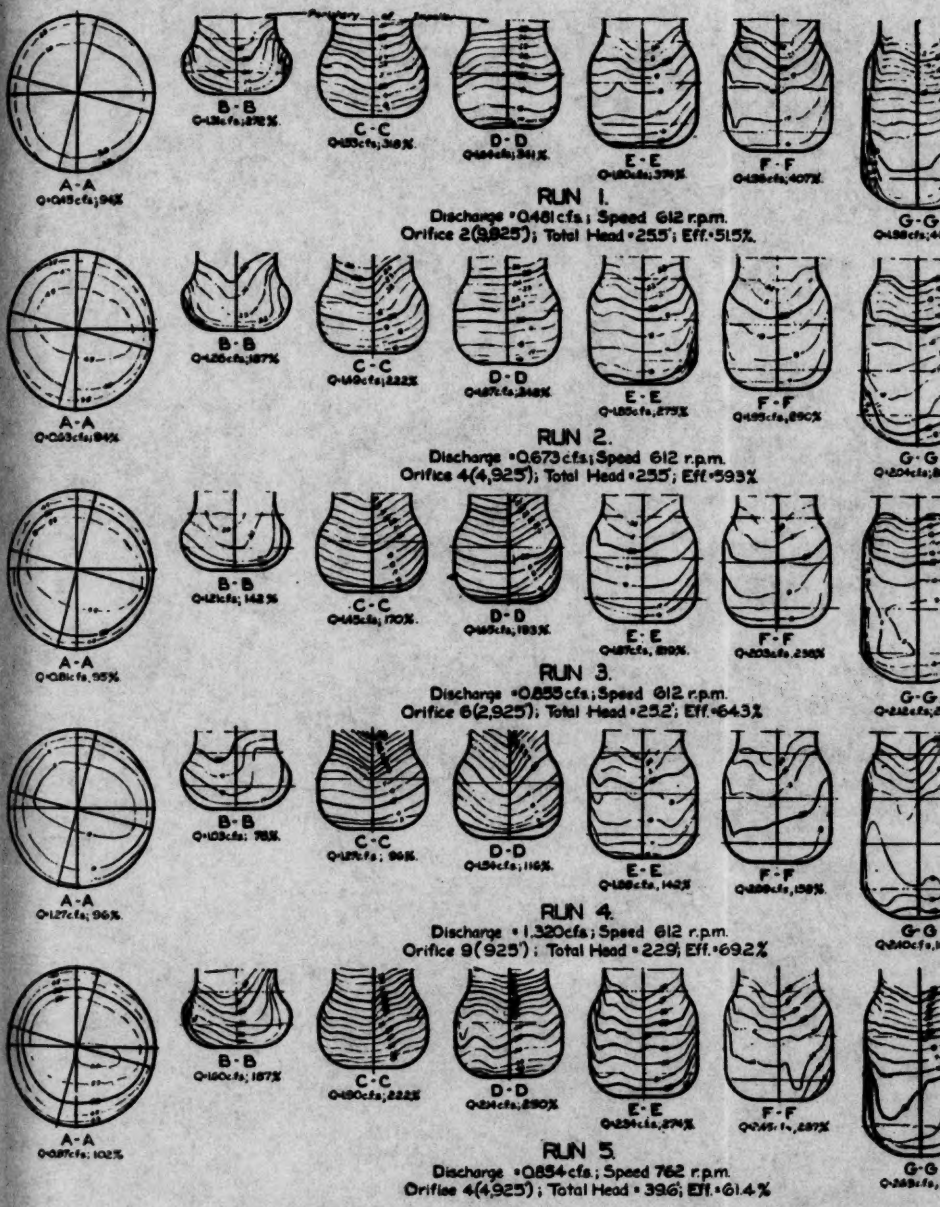
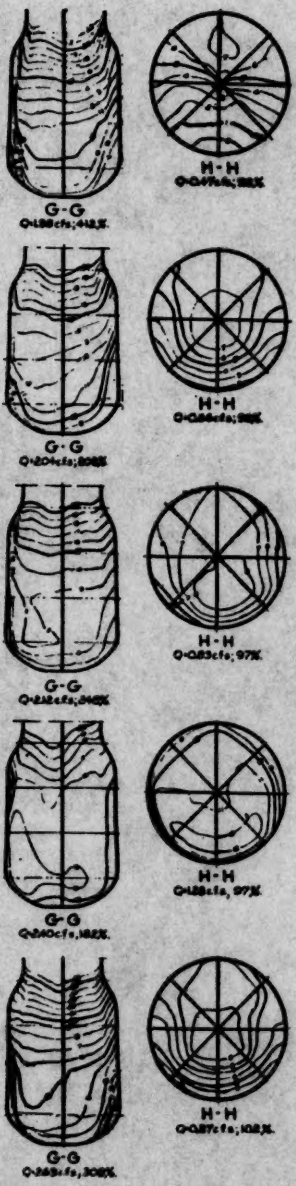
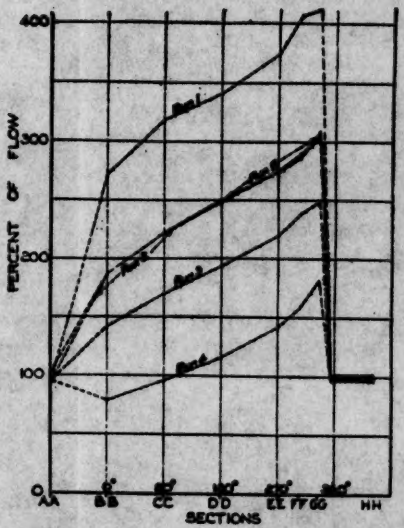
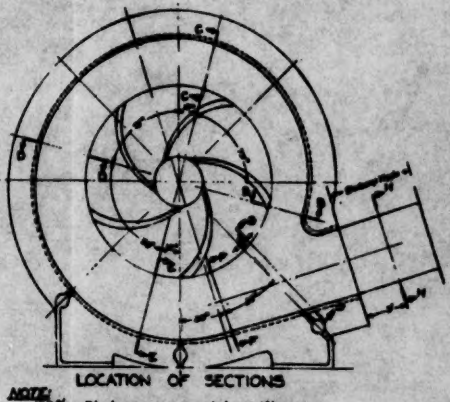

A-A
Q=0.48 cfs; 94%
B-B
Q=0.43 cfs; 87.2%
C-C
Q=0.53 cfs; 36.9%
D-D
Q=0.64 cfs; 36.1%
E-E
Q=0.80 cfs; 37.9%
F-F
Q=0.98 cfs; 40.7%
G-G
RUN 1.
Discharge =0.481 cfs; Speed 612 r.p.m.
Orifice 2(3,925'); Total Head =25.5'; Eff.=51.5%.
A-A
Q=0.63 cfs; 94%
B-B
Q=0.56 cfs; 187%
C-C
Q=0.48 cfs; 22.2%
D-D
Q=0.87 cfs; 24.8%
E-E
Q=1.05 cfs; 27.9%
F-F
Q=1.95 cfs; 89.0%
G-G
RUN 2.
Discharge =0.673 cfs; Speed 612 r.p.m.
Orifice 4(4,925'); Total Head =25.5'; Eff.=59.3%
A-A
Q=0.81 cfs; 95%
B-B
Q=0.71 cfs; 14.2%
C-C
Q=0.95 cfs; 17.0%
D-D
Q=1.65 cfs; 18.3%
E-E
Q=1.87 cfs; 21.0%
F-F
Q=2.03 cfs; 23.1%
G-G
RUN 3.
Discharge =0.855 cfs; Speed 612 r.p.m.
Orifice 6(2,925'); Total Head =25.2'; Eff.=64.3%
A-A
Q=1.27 cfs; 96%
B-B
Q=1.03 cfs; 7.8%
C-C
Q=1.27 cfs; 9.6%
D-D
Q=1.50 cfs; 11.6%
E-E
Q=1.86 cfs; 14.2%
F-F
Q=1.99 cfs; 15.9%
G-G
RUN 4.
Discharge =1.320 cfs; Speed 612 r.p.m.
Orifice 9(925'); Total Head =22.9'; Eff.=69.2%
A-A
Q=0.87 cfs; 10.2%
B-B
Q=1.60 cfs; 18.7%
C-C
Q=1.90 cfs; 22.2%
D-D
Q=2.14 cfs; 25.0%
E-E
Q=2.34 cfs; 27.4%
F-F
Q=2.45 cfs; 28.7%
G-G
RUN 5.
Discharge =0.854 cfs; Speed 762 r.p.m.
Orifice 4(4,925'); Total Head = 39.6'; Eff.=61.4%

FIGURE 4.







NOTE:
100% = Discharge measured by orifice.

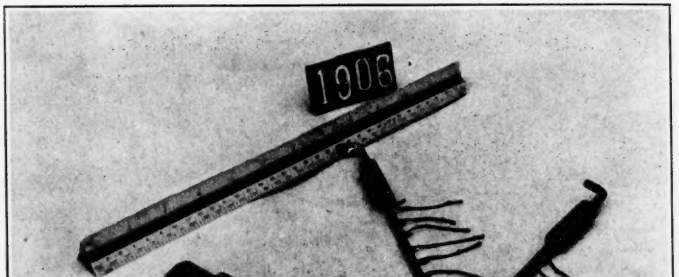


FIGURE 5.—A FEW OF THE FLOW DIRECTION INDICATORS USED FOR OBSERVATIONS IN THE VOLUTE AND NIPPLES.

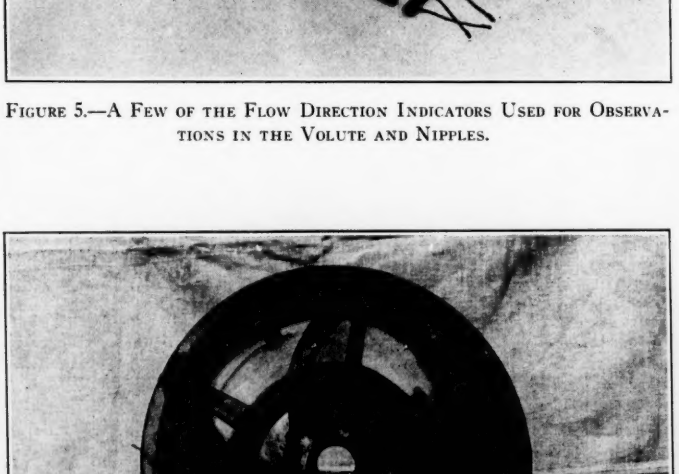


FIGURE 6.—FRONT VIEW OF IMPELLER "A" SHOWING LOCATION OF FLOW DIRECTION INDICATORS.

velocities are responsible for excessive costly wear. These factors will be further discussed in relation to efficiency and design features.

Pressure distributions were obtained for the operating conditions of the velocity experiments as stated for runs 2 to 5 of Figure 4. Pressures were measured at the inner surface of each opening provided for the velocity measurements and also at additional openings provided especially for pressures.

Observation indicated the existence of a pressure gradient along radial lines with a slope inverse to that of the velocity gradient as expected. Small losses of pressure head, especially noticeable between sections EE and FF, point to some unnecessary conversions to velocity head. No marked difference was noted between pressure distributions with Impellers "A" and "B".

It has been commonly believed that the pressures throughout the back clearance space are approximately equal to the pressures at the outlet of the impeller. The laboratory results obtained in this case have not substantiated the above belief; on the contrary, they have indicated that, at least for the conditions found on dredge pumps, the pressures in the back clearance space in the vicinity of of the axis of the pumps are much closer to those near the impeller eye than they are to the volute pressures.

Observation of Flow Conditions.

Direction indicators as shown in Figure 5 were used to study flow direction in the nipples and in the volute. These indicators consisted of stout silk threads fastened to streamlined brass rods which could be inserted or removed at will from the pump casing. Similar indicators were fastened in the interior of the impeller, between the shrouds as illustrated in Figure 6. These were removable also. Rubber balls with specific gravity slightly lower than that of water proved of assistance in studying flow conditions within the impeller and the volute. Other means of studying flow conditions within the volute, nipples, and clearance spaces such as air bubbles and permanganate solution were utilized.

The speed of rotation and the magnitude of the velocities of flow, especially within the impeller, were much too great for the human eye to follow. It was therefore necessary to use some means of arresting the motion. This was accomplished by means

of an electrical stroboscope. With the aid of this device, flashes of light may be obtained at any desired frequency. For this special application, the stroboscope was actuated by a mechanical contactor mounted on the dynamometer shaft, thus causing one flash of light for each revolution of the pump. Since the impeller was always illuminated in exactly the same angular position, it appeared to be standing still. Therefore, any movable object between the vanes which did not appear to stand still, gave an indication of the flow conditions or turbulence. Even for visual observations, conditions required the equipment to be powerful in effect on account of the great absorption of light by the total thickness of about 2 inches of pyralin and 4 inches of water.

In order to obtain permanent records of flow conditions within the pump, it was necessary to develop suitable photographic equipment for both still and motion pictures. A special ultra-high speed camera was coupled with a stroboscope of extreme power and frequency. Since the duration of a stroboscopic flash is in the order of 1/100,000 of a second, it will be readily understood that flashes of extreme intensity and actinic value had to be provided in order to obtain a satisfactory impression, even on the fastest emulsions available, and to be capable of doing this after having penetrated the highly absorptive pyralin and water.

These circumstances dictated extensive research and experimental work, involving the fastest lenses, fastest emulsions, and the most advanced methods of hypersensitizing and of development. A simplified form of stroboscopic circuit, developed by Professor H. E. Edgerton of the Massachusetts Institute of Technology, proved to be the most promising source of stroboscopic light. For this circuit, special tubes were designed and constructed for the purpose of increasing the intensity of the actinic light to the required degree. Figures 7, 8, 9 and 10 are typical of the results obtained with three stroboscope tubes flashing simultaneously. Professor Edgerton has estimated that the combined instantaneous active value corresponded to approximately 2,000,000 watts of incandescent lamps. The camera used for still photographs was a Leica fitted with a 73 mm. Hektor f : 1.9 lens using EK Super X film. Photographs were taken at a distance of approximately 34 inches with the lens wide open.

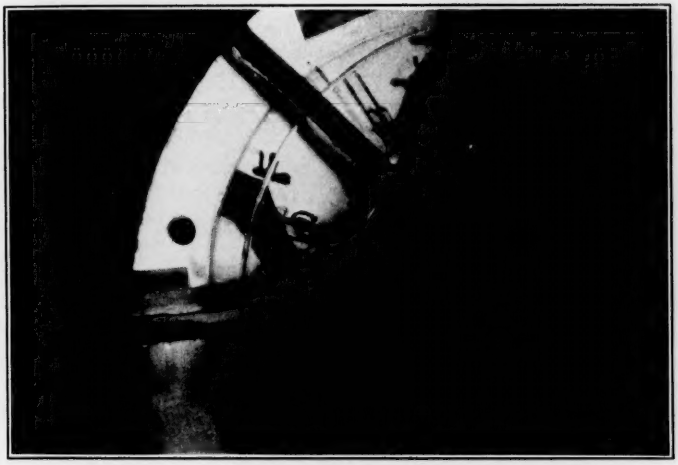


FIGURE 7.—FLOW DIRECTIONS WITHIN IMPELLER "A" USING ORIFICE 2 (DISCHARGE = 0.48 C. F. S.) AS PHOTOGRAPHED BETWEEN SECTIONS CC AND DD.

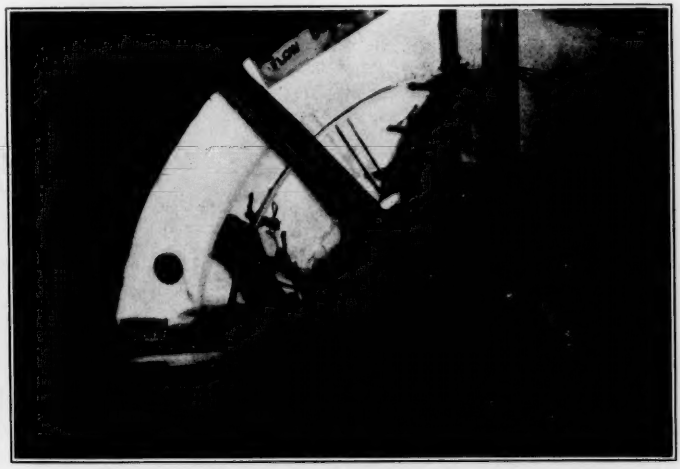


FIGURE 8.—FLOW DIRECTIONS WITHIN IMPELLER "A" USING ORIFICE 12 (DISCHARGE = 1.92 C. F. S.) AS PHOTOGRAPHED BETWEEN SECTIONS CC AND DD.

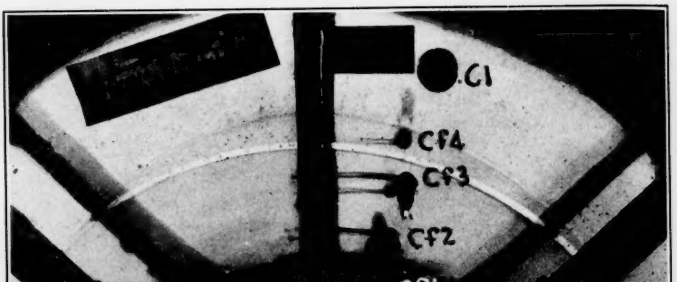


FIGURE 9.—FLOW DIRECTIONS BETWEEN FRONT HEAD AND FRONT SHROUD AT SECTION CC.

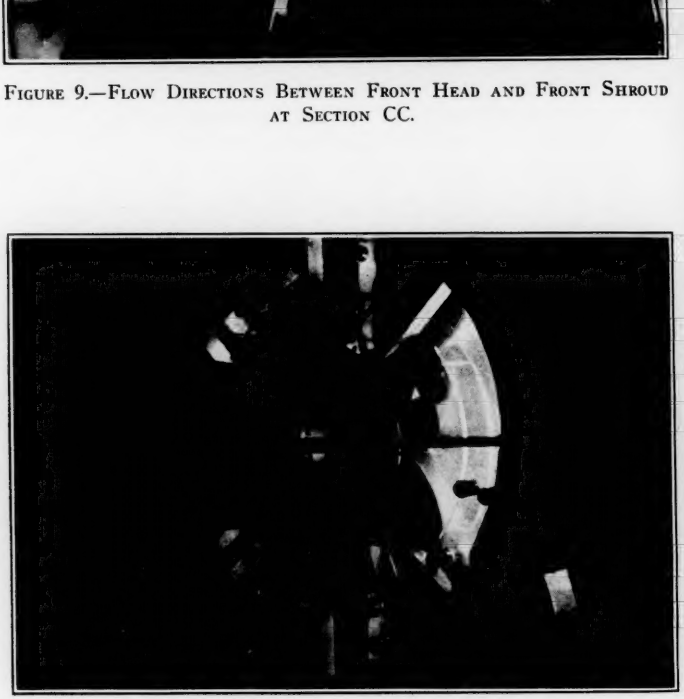


FIGURE 10.—RUBBER BALLS WITHIN IMPELLER "A" USING ORIFICE 6 (2925 FEET) AT 612 R.P.M.

Satisfactory moving pictures were obtained by synchronizing the camera with the stroboscope contactor, as well as with the pump rotation. This was accomplished by driving all three from the dynamometer shaft. The available camera in this case was a Cine-Kodak 16 mm. Special, equipped with a f : 1.9 Cooke lens and a shaft which controlled the shutter and film movement, so as to expose one frame of film on every complete revolution of the shaft. This camera shaft was driven by the dynamometer shaft through a long flexible cable. By synchronizing the flashes so as to occur when the shutter was wide open, one frame of film was exposed for every flash of light and thus recorded one of a long series of relative positions of the movable objects in the flowing water. With the pump rotating at 612 R.P.M., approximately 10 flashes and 10 exposures per second occurred. The stroboscope contactor was adjustable, permitting the flash and the exposure to occur at any desired point on the periphery.

Flow conditions were, generally, of a complicated nature and at times were subject to violent fluctuations and changes, making an accurate description difficult.

The average directions of flow in the volute were obtained for Impellers " A " and " B " using orifice 6 at 612 R.P.M. by observation of the threads used as indicators. The general flow direction suggested a clockwise spiralling of the stream to an observer looking in the same direction as the flow occurred. The individual thread indicators fluctuated rapidly over ranges varying from 15 to 45 degrees. However, no striking differences were noticed between the two impellers. With both impellers the most severe fluctuations occurred in section BB. Rubber balls sent through the pump frequently struck the tongue and were often observed to travel around the volute several times before entering the discharge nipple.

Figure 9, taken of section CC (top of front head of pump) shows three pairs of threads directed radially inward, indicating a fairly strong flow through the clearance space toward the center of the pump. Observation of the flow through the clearance space at the back head of the pump resulted in like conclusions. In both cases, these conclusions were verified with injection of permanganate solution into both clearance spaces.

Figures 7 and 8 illustrate typical flow directions within Impeller "A" for the two extremes of discharge indicated. The location of the threads point out the changes that occurred within the impeller as the discharge was increased by shortening the pipe lines. The presence of fluctuations and vibrations of the thread indicators, which are especially violent with small discharges, is not apparent from the photographs. In the cases just cited, the indicators were located ½-inch away from the vane surface. A second group of indicator rods was spaced at the third intervals between the vanes. These also demonstrated the gradual improvement in the flow conditions as the discharge increased. Visual observation of the flow directions within Impeller "B" brought forth no striking differences in the flow conditions of the two impellers.

The Experiment with a Tongue Extension.

By radically reducing the minimum area of the volute near the tongue, it was hoped to bring about an improvement in the efficiency as a result of an extensive reduction in the volume of recirculated water. The reduction in area was accomplished by inserting a rubber extension to the tip of the tongue as shown in Figure 11.

The pump characteristic with the tongue extension in place was determined for a model speed of 612 R.P.M. Figure 11 represents a comparison of this characteristic with that of the pump under same operating conditions, but without tongue extension. It will be noted that for the condition with tongue extension both heads and efficiencies obtained were decidedly lower throughout the range. This striking result indicates that the turbulence losses exceeded any possible gains due to the reduction of the quantity termed recirculating flow. It is observed that tongue extension causes less change in characteristic for large discharges than for small ones; this seems reasonable, considering the fact that the tendency toward recirculating flow is much greater with small discharges.

The action of rubber balls was studied by continuous and stroboscopic light for orifice 6 (normal discharge). From this study it was concluded that the tendency to remain within the impeller was approximately the same as without the extension. A large

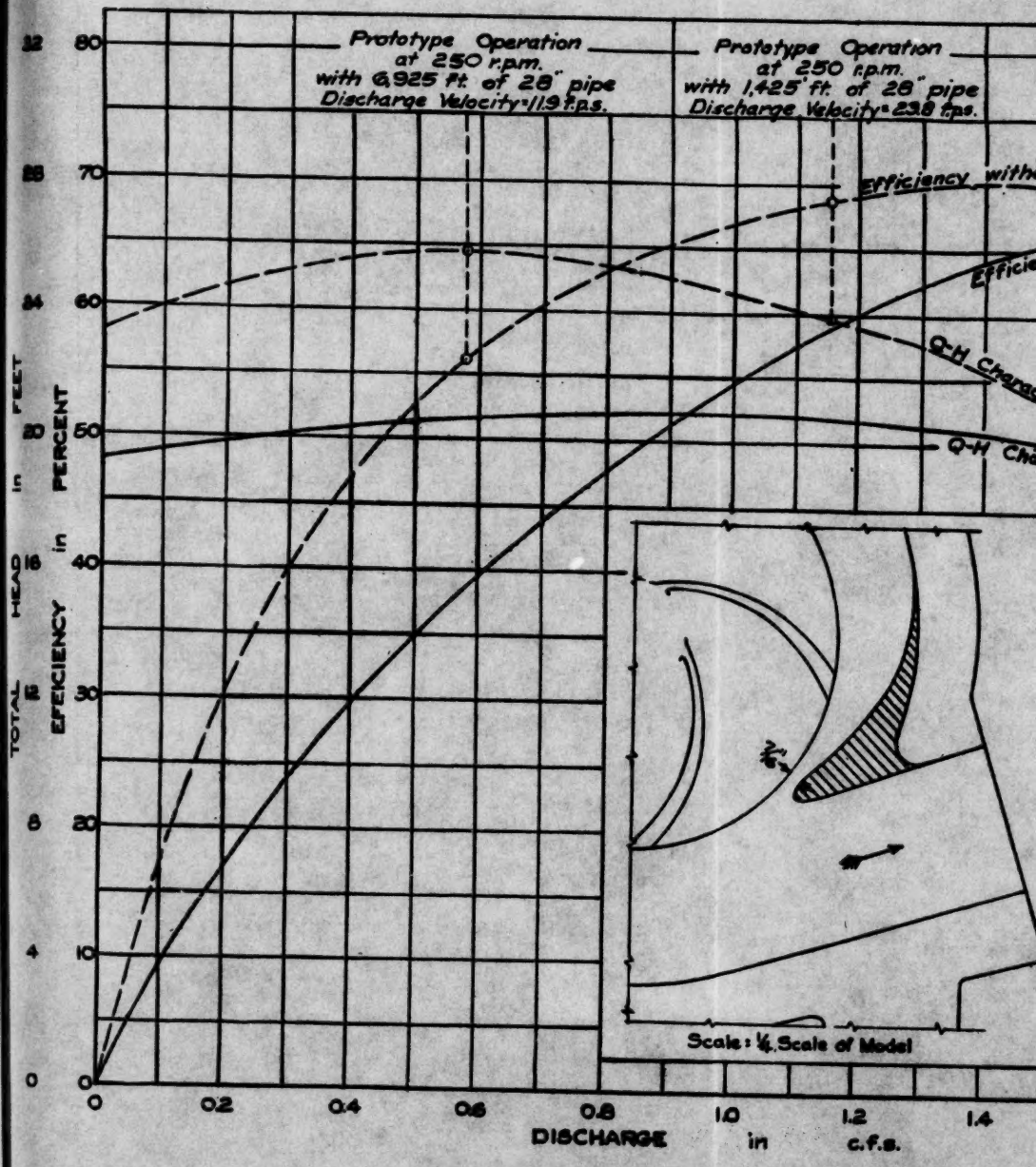
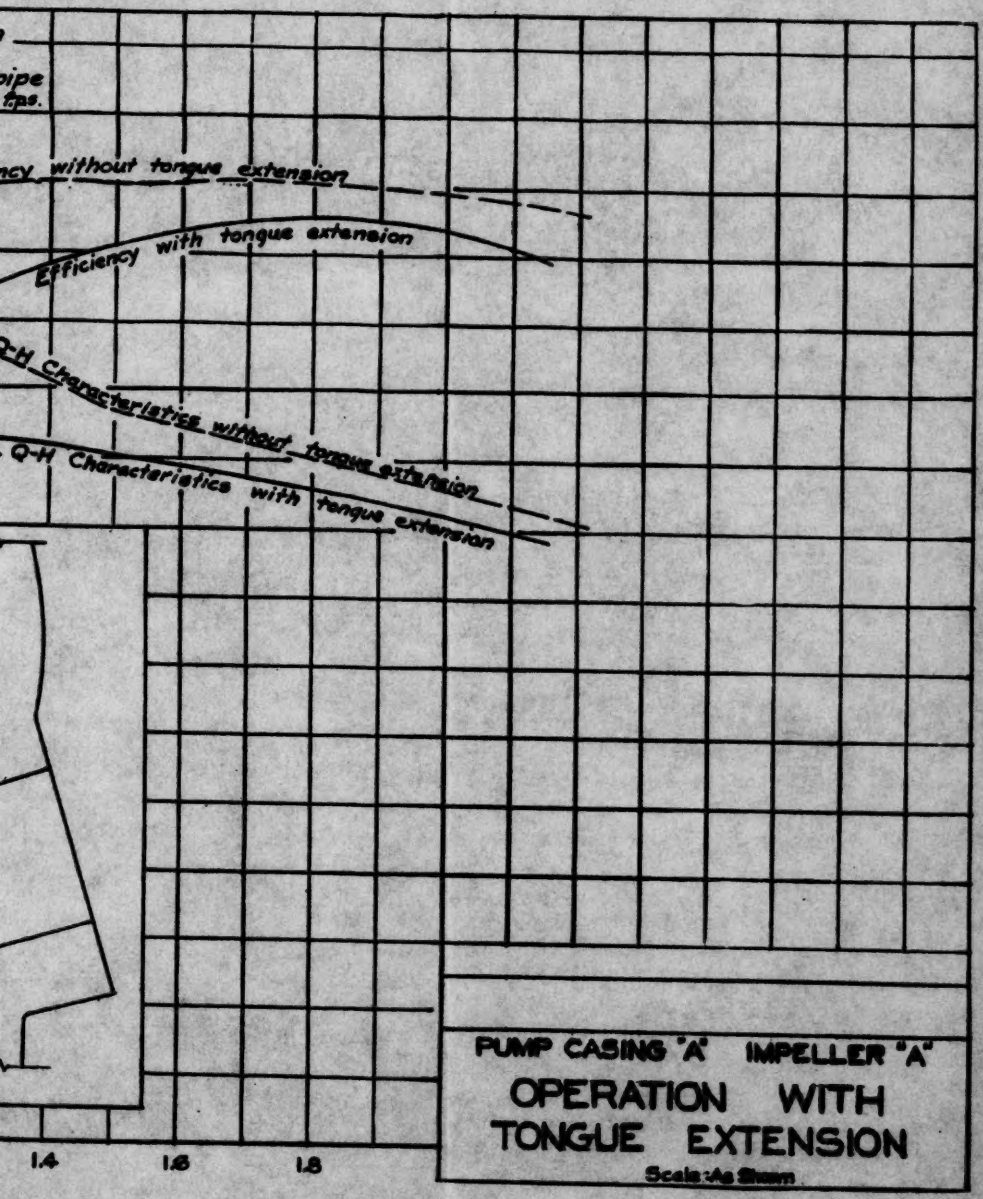

Prototype Operation at 250 r.p.m. with 6,925 ft. of 28" pipe Discharge Velocity = 11.9 f.p.s.
Prototype Operation at 250 r.p.m. with 1,425 ft. of 28" pipe Discharge Velocity = 23.8 f.p.s.
Efficiency with
Effici
Q-H Chara
Q-H Cha
TOTAL HEAD in FEET
EFFICIENCY in PERCENT
DISCHARGE in c.f.s.
0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
Scale: ¼ Scale of Model

FIGURE 11.


ncy without tongue extension
Efficiency with tongue extension
Q-H Characteristics without tongue extension
Q-H Characteristics with tongue extension
1.4
1.6
1.8
PUMP CASING "A" IMPELLER "A"
OPERATION WITH TONGUE EXTENSION
Scale: As Shown

1.

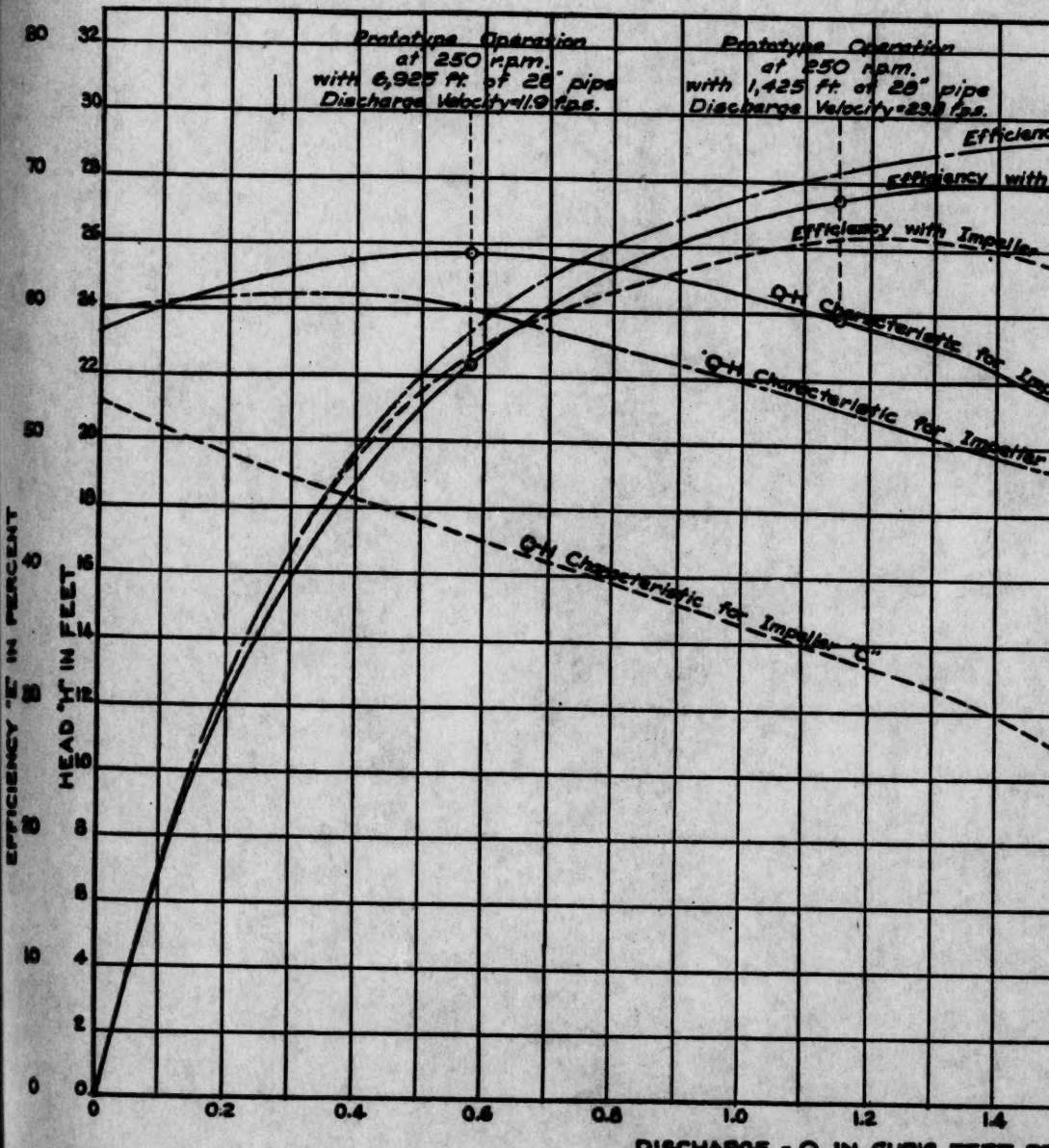

Prototype Operation at 250 r.p.m. with 6,925 ft. of 28" pipe Discharge Velocity=11.9 f.p.s.
Prototype Operation at 250 r.p.m. with 1,425 ft. of 28" pipe Discharge Velocity=23.8 f.p.s.
Efficiency with Impeller
Efficiency with Impeller
Q-H Characteristic for Impeller
Q-H Characteristic for Impeller
Q-H Characteristic for Impeller "C"
EFFICIENCY "E" IN PERCENT
HEAD "H" IN FEET
DISCHARGE - Q IN CUBIC FEET PE
0
0.2
0.4
0.6
0.8
1.0
1.2
1.4

FIGURE 12.

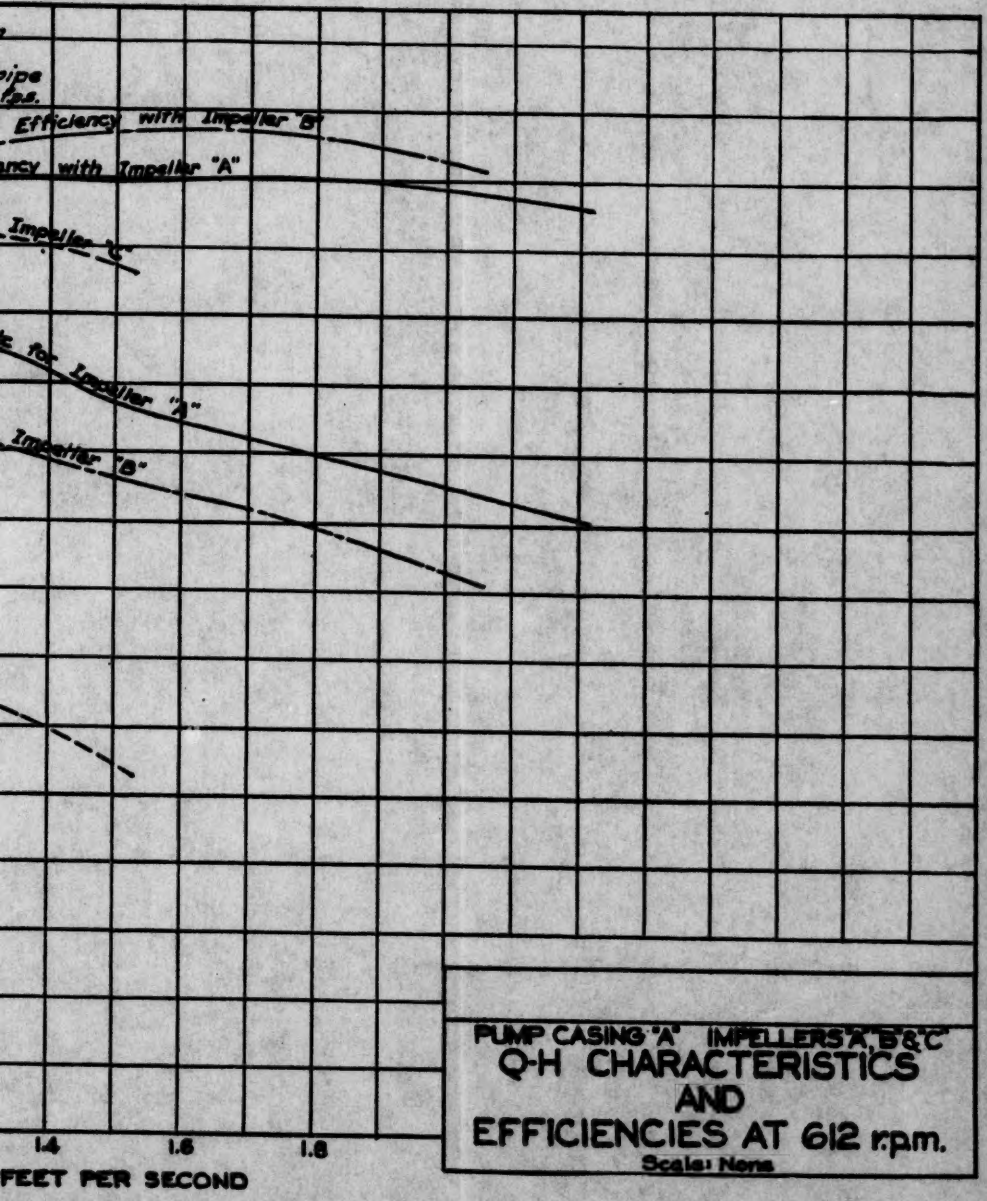

Efficiency with Impeller "B"
ncy with Impeller "A"
Impeller "C"
for Impeller "A"
Impeller "B"
1.4
1.6
1.8
FEET PER SECOND
PUMP CASING "A" IMPELLERS A, B & C
Q-H CHARACTERISTICS
AND
EFFICIENCIES AT 612 r.p.m.
Scale: None

FIGURE 12.

percent
withsta
the ext
and the
BB an
eddies
of the
through
turvatu
drop in

Data
model
actual
speed,
factor
based
laws w
speed
As the
it varie
horsepo
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ering th
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"B",
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The sl
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impelle
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slightly
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percentage of the balls was still recirculating in the volute notwithstanding the presence of the extension. In order to pass by the extension, the balls re-entered the impeller between section FF and the tongue and then returned to the volute between sections BB and CC. With the aid of permanganate solution, strong eddies and extreme turbulence was observed in the front half of the volute after the flow passed the tongue extension, while through the rear half of the volute the flow was parallel to the curvature. This observation substantiated the reasons for the drop in efficiency stated above.

Factors Influencing Efficiency.

Data from which efficiencies were computed were obtained for model speeds ranging between 250 and 980 R.P.M. Since the actual observations could not always be made at exactly the desired speed, it was therefore necessary to make adjustments for this factor before making any comparisons. These adjustments were based on the fundamental principles of pump design or affinity laws which are stated below. The discharge Q varies with the speed N, and the head H varies with the square of the speed N. As the water horsepower WHP is a function of both Q and H, it varies as the cube of the speed N. It is assumed that brake horsepower BHP also varies with the cube of the speed N. Though not strictly true, this results in no appreciable error when considering the small variations in speed involved.

The efficiency data obtained at 612 R.P.M. with impellers "A", "B", and "C", are shown graphically in Figure 12. The operating conditions were practically identical so that their influence may be neglected when comparing the different designs. The slight irregularity in the QH characteristic for impeller "A" does not exist with impeller "B". The rising characteristic and large "hump" in the QH characteristic of impeller "A" may be conducive to less favorable operation than for impeller "B" under certain conditions. Impeller "B" operates at a definitely higher efficiency than impeller "A"; however, impeller "B" has the disadvantage that it must be operated at a slightly higher speed to obtain the same discharge through a given pipe line. The efficiency of Impeller "C" is definitely below

that of the other impellers for high discharges and therefore would require a much higher speed for a given condition of operation.

The influence of mechanical losses on the values of brake horsepower and efficiency is much more pronounced in the model than in the prototype. This is especially true at low speeds and discharges, accounting essentially for the low model efficiencies at low speeds.

The effect of increasing the speed or discharge is to reduce the percentage of water horsepower which is lost due to hydraulic friction. This trend is apparent in both model and prototype data though to a different degree in each.

The influence of suction head as indicated by the model data was considerably greater than expected. For vacuums of 10 inches of mercury or less or positive inlet pressures, the effect of suction head on efficiency was not measurable. However, for suction heads greater than 10 inches of mercury, a large quantity of air appeared in the pump and the efficiency became increasingly dependent upon suction head, although most of the data were for vacuums much too low to cause cavitation. This phenomenon was no doubt partially a result of model construction, as for high suction heads it was extremely difficult to eliminate small air leaks in the numerous joints between metal and pyralin. The jet through the measuring orifice discharged directly into the atmosphere, creating conditions favorable to air saturation of the water. Also the presence of a check valve with rather high constriction at the entrance of the suction line caused a sudden drop in pressure at that point. It is believed that these last two factors affected the results of high vacuum comparisons on account of excessive liberation of air out of solution.

An analysis was made of the effect of flow distribution and recirculation upon efficiency with impeller "A". The average velocity through each section was determined by measuring with a planimeter the area between each pair of iso-velocity lines on the original full scale drawings of Figure 4. These data were then used to compute the flow through each section, both in C.F.S. and percentage of measured pump discharge. To do this, it was necessary to choose a point from which it was assumed that all flow from the impeller at that point would circulate around the volute a full 360 degrees. This was done by extending the upper

line of the discharge nipple to meet the periphery of the impeller. This occurred approximately at section FF. If there were no recirculation in the pump, the flow post section FF would be equal to the measured pump discharge. Thus the quantity of flow that may be charged to recirculation is equal to the difference between the computed flow post section FF and the pump discharge measured by the orifice. It is realized that these assumptions are somewhat arbitrary as it was not possible to definitely determine the point at which the discharge from the impeller flowed directly into the discharge stream. It is also probable that the position of this point varies with the pump discharge.

As illustrated in Figure 4, runs 1, 2, 3, and 4, are for the same speed of 612 R.P.M. but with successively larger discharges. The pump discharge measured by the orifice is taken as 100 per cent. In sections BB and CC it will be noticed that as the pump discharge increased, the flow through these sections decreased indicating a rapidly decreasing percentage of recirculating water. The highest efficiency recorded for the velocity experiments was obtained with the largest of the 4 orifices corresponding to the shortest pipe line. The flow conditions for this orifice are the most uniform of the four runs. The table below demonstrates the effect of unfavorable working conditions of the longer pipe lines and suggests the possibilities of further improvements in design with the purpose of improving flow conditions and quantitatively reducing recirculation and thereby obtaining higher efficiencies.

	Flow Through BB in per cent of Pump Discharge	Pump Efficiency in per cent
Run 1 with orifice 2 (9925 feet)	Per cent 272	Per cent 51.5
Run 2 with orifice 4 (4925 feet)	187	59.3
Run 3 with orifice 6 (2925 feet)	142	64.3
Run 4 with orifice 9 (925 feet)	78	69.2

Runs 3 and 5 were made with the same pump discharge at different speeds on account of the different pipe line lengths used. (See Figure 4.) From these it is concluded that the effect of speed upon the velocity distributions in the nipples is practically negligible. Due to the higher impeller speed, the average velocities in the volute increased, thus increasing the percentage of recirculating water from 142 per cent to 187 per cent. A corresponding drop in efficiency from 64.3 per cent to 61.4 per cent occurred.

A comparison of flow conditions within the impeller and the efficiency data leads to the logical assumption that the extent of deadwater zones within an impeller has a pronounced influence on the relative efficiency of impellers. A similar conclusion was reached by Taijiro Kasai Kogakushi after a study of the exit velocities and slip coefficient in centrifugal pumps. He states "another interesting fact can be found about the characteristic curves; that the greater the size of the deadwater zones in the impeller passages at the discharge for N_{max}, the lower the velocity of N_{max} becomes." N_{max} is the maximum efficiency obtainable from a given design at a given speed.

Similitude and the Acting Forces.

Complete geometric and dynamic similitude between any model and its prototype occurs only when all the forces acting bear the same ratio to each other at any given instant. Since inertia forces are always present, dynamic similitude is dependent upon the nature of the other forces acting.

The fundamental relations between H, Q, and N which govern the performance of a centrifugal pump depend on the accelerative forces acting on the impeller and upon the geometric design. If elastic forces are held sufficiently low, similitude between these relations in model and prototype will exist regardless of model speed. This, simply stated, is Newton's law of motion or similitude ($F = Ma$). Froude's, Cauchy's, and Reynolds' laws, which are merely special cases of this, will be discussed individually.

Terrestrial Gravitation Forces or Froude's Law of Similitude.

Cases in which these forces are considered predominant require the application of Froude's law of similitude. This, stated mathematically, is

$$\frac{V^2}{Lg} = \text{constant.}$$

This relation was derived in the following manner:

From Newton's law, $F = \frac{d(mV),}{dt}$ the inertia forces are proportional to $\rho \frac{L^3V}{T} = \rho L^2 \left(\frac{L}{T}\right)V = \rho L^2V^2$

where L is any linear dimension.

The force of gravity if proportional to $wL^3 = g\rho L^3$

Froude's law of similitude stated mathematically is merely a constant ratio of these forces or

$$\frac{\text{inertia forces}}{\text{force of gravity}} = \frac{\rho L^2V^2}{g\rho L^3} = \frac{V^2}{gL} = \text{constant}$$

V = velocity

ρ = density

g = Acceleration of gravity

F = force

If the subscripts m and p be used to indicate model and prototype values respectively, and be used to represent the scale ratio, then:

$$\lambda = \frac{L_p}{L_m}$$

Therefore to observe Froude's law

$$\frac{V^2_m}{gL_m} = \frac{V^2_p}{gL_p}$$

Since $V = LN$, $\frac{L^2_mN^2_m}{L_m} = \frac{L^2_pN^2_p}{L_p}$ or

$$\left(\frac{N_m}{N_p}\right)^2 = \frac{L_p}{L_m} = \lambda$$

From which $N_m = \lambda^{1/2}N_p$

Therefore, the speed required for a 1:6 scale model to represent a prototype speed of 250 R.P.M. is

$N_m = 2.45 \times 250 = 612$ R.P.M. as indicated in the section discussing the simulation of prototype pipe line lengths.

Elastic Forces or Cauchy's Law of Similitude.

Elastic forces influence the performance of a centrifugal pump operating under conditions of high vacuum involving cavitation or air in the system. Consideration of these forces requires the observance of the so-called Cauchy Law of Similitude, which stated mathematically is $\frac{V^2\rho}{E} = \text{constant}$.

This may be derived as follows:

From the fundamental expression relating the direct load in tension or compression to (elongation or shortening) and elasticity we have

$$e = \frac{Pl}{AE} \text{ or } P = \frac{eAE}{l} = \frac{LL^2E}{L} = L^2E$$

Also: $\frac{\text{inertia forces}}{\text{elastic forces}} = \frac{L^2V^2\rho}{L^2E} = \frac{V^2\rho}{E} = \text{constant}.$

e = elongation or shortening
E = modulus of elasticity
ρ = density
V = velocity
L = any linear dimension
$A = L^2$ = area
P = force
l = length in the direction of application of the force P

Using subscripts as before,

$$\frac{V^2_m\rho}{E} = \frac{V^2_p\rho}{E} \text{ or } V^2_m = V^2_p \text{ for the same liquid}$$

$$\therefore L^2_m N^2_m = L^2_p N^2_p \text{ or } N_m = \frac{L_p N_p}{L_m} = \lambda N_p$$

The speed required for a 1:6 scale model to represent the prototype is therefore

$$N_m = 6 \times 250 = 1500 \text{ R.P.M.}$$

This is the condition normally used in pump model tests as it likewise produces equality of heads, velocities and stresses between model and prototype. High vacuum model data are not transferable to prototype nor can prototype operation with high vacuums be predicted unless this law is satisfied.

The Consideration of Viscous Forces or Reynolds Law of Similitude.

From fundamental relations it is seen that the friction force F equals $\frac{\mu AV}{y}$ where μ is the absolute viscosity of the fluid and y is any linear distance from a surface to which the flow is relative to a plane at which the velocity V is in effect. A is the area over which the force F acts. Expressing y and A in terms of L we have $F = \frac{\mu L^2 V}{L}$ and hence is proportioned to μLV. Therefore the ratio of inertia forces to friction or viscous forces is proportional to $\frac{L^2V^2\rho}{LV\mu} = \frac{LV\rho}{\mu} = \frac{LV}{\gamma} =$ constant as stated for Reynold's Law of similitude. $\gamma = \mu/\rho$ and is known as the kinematic viscosity of the fluid.

Assuming the same liquid (water) to be used for both model and prototype and using the same subscripts as in the two previous cases we have

$$L_m L_m N_m = L_p L_p N_p$$

$$\text{of } N_m = \left(\frac{L_p}{L_m}\right)^2 N_p = \lambda^2 N_p$$

Therefore, the speed required for a 1:6 scale model to represent a prototype speed of 250 R.P.M. is

$$N_m = (6)^2 \times 250 = 9000 \text{ R.P.M.}$$

While such speeds are obviously impractical for turbine or pump models, in their absence some discrepancy of brake horsepower and efficiency may be expected. In the field of turbine testing the practiced procedure has been the development of an empirical correction for the losses as in the Moody formula.

General Conclusions.

Outstanding phenomena, brought to light as a result of this visual investigation, are the existence of large deadwater areas within the impeller and the consequent existence of extremely high velocities over a small percentage of the cross-sectional area within the impeller. The magnitude of these velocities is often 6 times the average for the entire area. On account of these extreme velocities, considerable energy losses occur as a result of the transformation from potential to kinetic energy and vice versa. It is believed that these losses may be materially reduced by changes in design after further experimental effort.

The phenomena of dead water areas and maldistribution of velocities are also existant in conjunction with fluid flow through bends in pipes and conduits resulting in extensive energy losses. These influences must be detrimental to the transfer of fluid through the various piping systems of naval vessels where space is at a premium and elbows, manifolds, and bends are in abundance. The decrease in power requirements and improvement in efficiency of these systems could well offset the cost of further experimental effort in this field.

Many aspects of a broad and rudimentary subject have been discussed in brief. If this paper succeeds in raising more questions than providing answers, thereby stimulating further investigation, the effort expended in its preparation will have been well repaid.

The author wishes to thank Henry L. Setz, Mechanical Engineer with the Bureau of Ships, for his assistance in editing and assembling the data for this paper. His knowledge of mathematics was very valuable especially in the interpolation of the various laws of Similitude.

TODAY'S MACHINE TOOLS, MACHINERY AND MECHANIZED WAR.

Lt. Commander R. E. W. Harrison, U.S.N.R.

It seems that we must learn our most vital lessons in the hardest way. That the greatest machine tool producing country in the world should find itself confronted with a threat to its very existence and at the same time gradually wake up to the fact that it is short of the very items which made its previous prosperous existence possible is one of those extraordinary things which bear out the truism that fact is stranger than fiction. That America, which has lead the world for thirty years in the conception and quality of its machine tools, should enter a major war emergency, deficient in these tools, is a pathetic indictment of those whose leadership could and should have saved us from this condition.

In the first world war England, France, America and Italy developed the technique which today we know as mechanization of war and poor though by comparison they were, the British tanks at Cambrai turned the tide—the German tanks were a failure. The vital significance of this, however, was that gone forever was the old technique of war which treated in terms of horse flesh, human sweat, and 30-caliber bullets; the age of mechanization was on us and armies measured their mechanical equipment in hundreds of thousands of tons of mechanization instead of in so many head of cattle and so many effective bayonets.

In the hands of her land forces the effective use of the bayonet wrote the history of the British Empire for a hundred years. The retention of this tool today is little more effective than the officer's saber, and even its most useful function as a toasting fork is disappearing with the provision of more adequately mechanized field kitchens.

Machinery and Tools as Weapons of War.

It is recalled that in 1917 a Colonel of U. S. Ordnance spent four months of his valuable time scouring the United States in an

endeavor to procure a simple standard engine lathe, so scarce were these basic tools of mechanization at that time.

When Germany's Nazi Party hatched its plans for European and subsequent world domination it was quick to put into effect policies embodying the lessons of World War No. 1. The first of these lessons was that mechanization was here to stay and could only be provided by an absolutely adequate supply of machine tools. In 1934-35 the German government offered a premium to all actual and potential munitions making factories by subsidizing the acquisition of new, up-to-date, and highly productive tools and plant equipment. The plan permitted the manufacturer to amortize the tools in twelve months instead of the normal 240 months and the policy was so effective that, whereas it was intended originally to put it into effect for one year, it was extended for a second year until the whole of basic German industry in the mechanical field was carried on a solid unshakable foundation of most efficient machinery, tools, and practices. In this one policy alone Germany laid the fundamental basis for the blitz that she was preparing for the whole world.

How effective this planning has been the democracies know only too well. Their inadequate weapons, and their out-worn tactics have had to be dropped or modernized under the unhappy conditions of actual active war and the defense of security paid for in casualties and deepest misery. There are endless pictures, and stories without number, of groups of British and French troops who stood in the paths of the Panzer divisions opposing with their thirty-caliber rifles and puny bayonets, modern tanks, so well equipped with armor that nothing less than a 75 mm. direct hit could make a dent in them. The crass stupidity of the military mind which permitted its people to go into battle with little more than their courage and a willingness to die forms a bitter and costly indictment of the experts who did not hammer home their story of what the Germans were doing, and the politicians who would not listen to them and profit by the reports of their accredited military and naval attaches.

By 1937 German factories for the production of tanks, machine guns, airplanes and cannon as well as motor vehicles, combat cars, field kitchens and airplane bombs were equipped with modern machinery and tools which would permit them to turn out their

products with a speed and precision heretofore completely unknown anywhere in this world.

What was happening to America and in a measure to England and France, the three great democracies, in the meantime? In England the industrial giant was just beginning to stir again from a sleep of industrial depression reminiscent of "Rip Van Winkle." In France a more wearied people had turned to a form of state socialism which manifested itself in shorter hours, higher wages, unearned labor privileges and costs which economic rehabilitation was totally unable to sustain. In America we were recovering from 1932, that dreadful economic calamity which also was a legacy of the world war. Most of our machinery and equipment, *i.e.*, about 65 per cent of it was much over ten years old and financial forecasts were such that investors balked at investing their savings in new and up-to-date machinery. Despite the fact that we were the most highly mechanized people in the world, in so far as the amount of mechanization per capita was concerned, much of our equipment was out of date and its very obsolescence made much of it more of a burden than an asset. Mechanization in our democracy was gaudy in size but relatively feeble in effectiveness, so feeble in fact that its productive efficiency had extreme difficulty in offsetting the cheaper labor of competing countries seeking a share of the world market. The only one really notable exception was the automobile industry.

Discussions between industrialists and the government of the U. S. at the time when Hitler was rehabilitating the German basic industries brought forth nothing but opposition to any inductive plan of accelerated mechanization. In fact, the N.R.A. was evolved to regulate industry and competition between these shabbily equipped industries and N.R.A.'s net result being monopolistic, it eventually toppled of its own weight.

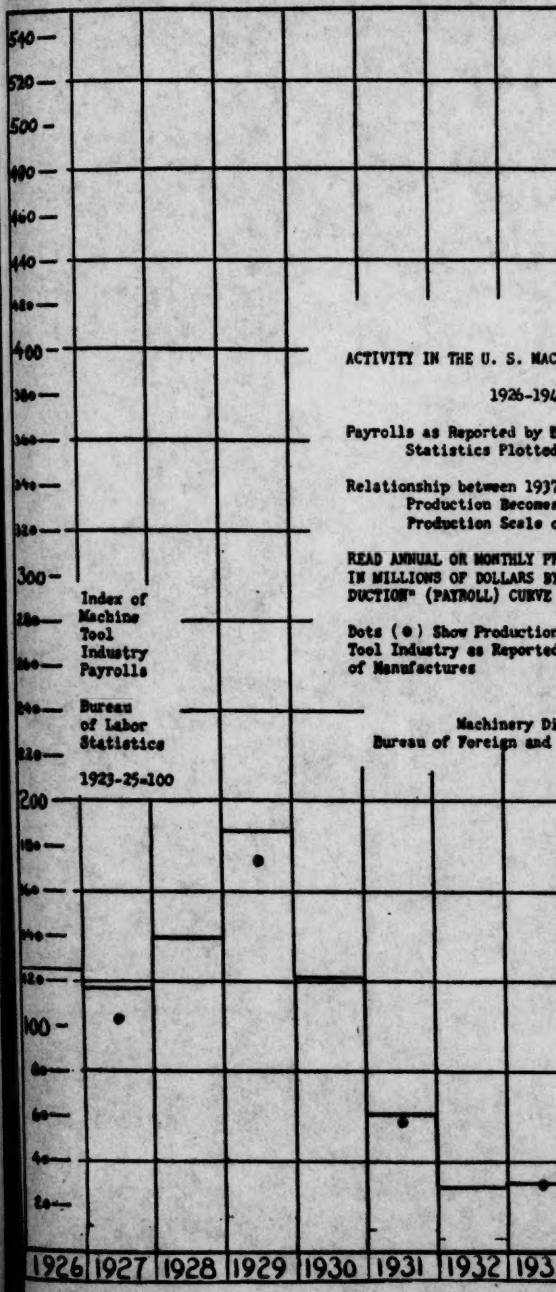
While Germany was re-equipping its industries aided by accelerated depreciation allowed as deduction from income tax, the policy of our Federal Government was directed in a diametrically opposite channel. First, was N.R.A. which divided industry into groups and erected protective economic umbrellas over the least efficient in the groups. No more effective plan of perpetuating obsolescence could have been conceived than the price fixing powers which were assigned to N.R.A. by Congress. N.R.A.'s defeat and

elimination were largely brought about by competitive business interests believing in competitive economy and profit. None can forget the feeling of relief which was experienced when the Supreme Court delivered its verdict over the puny body of a very dead chicken.

As though this were not enough, Congress then proposed further retribution, along exactly opposite lines to those used by Germany, by its proposal to increase Federal revenues by voting a reduction in depreciation allowances for tax purposes.

As a result of discussions between the Treasury Department and Congress, the Treasury promised to reduce depreciation rates throughout industry and to secure extra income without legislation. Concurrently, a minority of manufacturers in the U. S. who had carefully weighed the effects of the German armament acceleration due to the depreciation plan, became vocal. Eventually the discussion degenerated into a political free-for-all as those in Congress who favored the setting of obsolescence or depreciation rates by fiat grabbed at the proposal for consideration of the German plan in the belief that it would bolster their case, *i.e.*: that depreciation could be made a matter of law irrespective of what the facts of the case might happen to be. This of course was absolutely contrary to the previous U. S. conception of a democratic economy and much time and thought were devoted by another little band of "rugged individualists" who fought tooth and nail with all concerned to maintain that industrial cannabalism is a necessary concomitant of free enterprise and that the rate at which the industrial Moloch wore out its teeth was a matter of industrial progressiveness and efficiency, and that it was known and evaluated by these conditions only, lawmakers notwithstanding.

It is worthy of note at this time, that income taxes started with World War No. 1. Prior to that time the practice had been to write off capital investments as rapidly as possible, which course placed the manufacturing companies in excellent position to take advantage of every item of modernization offered for their production lines. Fundamentally, the policy of the Treasury Department in placing the onus of proof of depreciation on manufacturers, thereby catching about 95 per cent of them unprepared with proof, placed a large, thick, damp, suffocating blanket upon


ACTIVITY IN THE U. S. MAC
1926-194
Payrolls as Reported by B
Statistics Plotted
Relationship between 193
Production Become
Production Scale
READ ANNUAL OR MONTHLY FI
IN MILLIONS OF DOLLARS BY
DUCTION" (PAYROLL) CURVE
Dots (●) Show Production
Tool Industry as Reported
of Manufactures
Machinery Di
Bureau of Foreign and
Index of Machine Tool Industry Payrolls
Bureau of Labor Statistics
1923-25=100
540
520
500
480
460
440
420
400
380
360
340
320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
1926
1927
1928
1929
1930
1931
1932
193

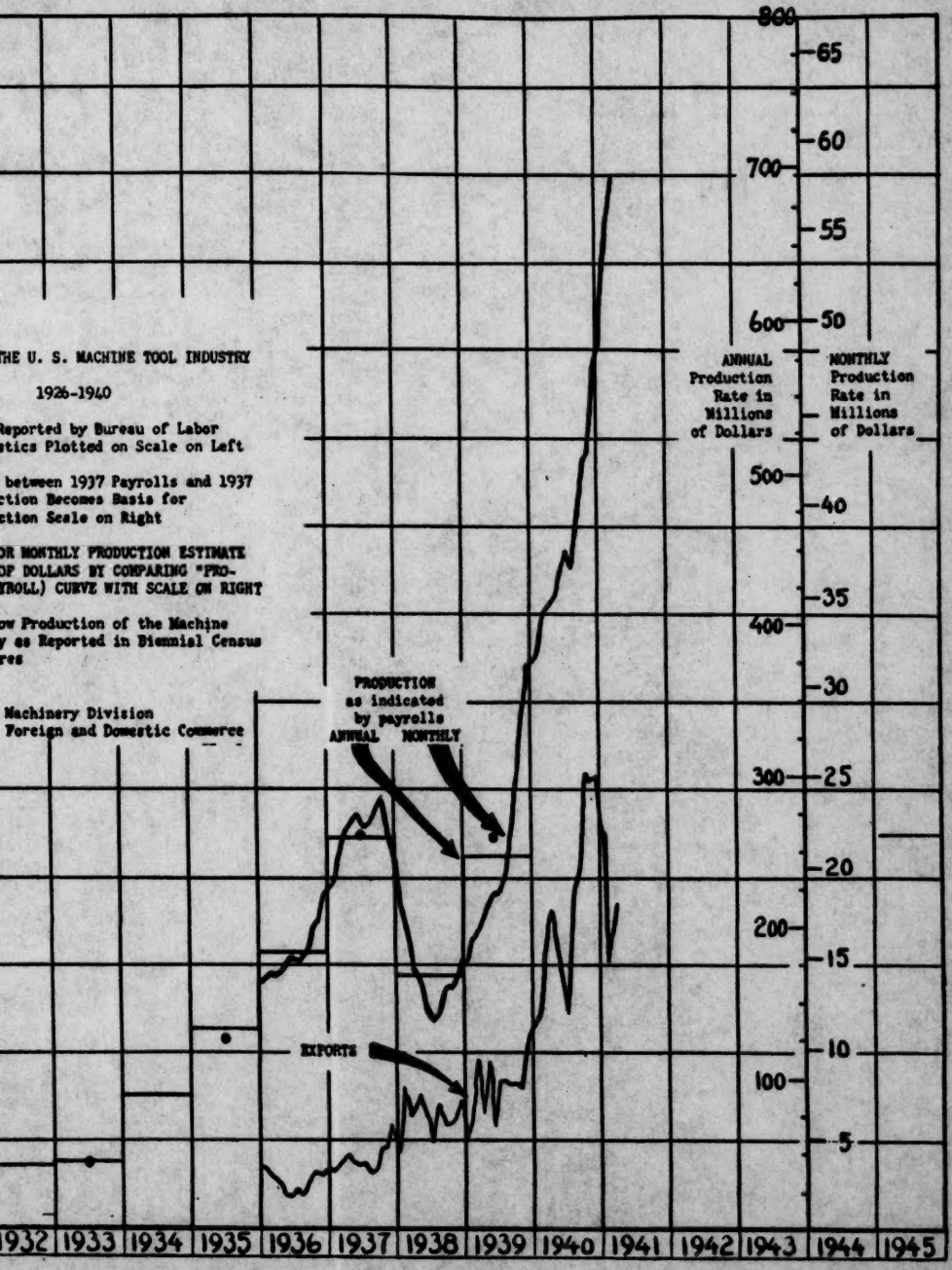

THE U. S. MACHINE TOOL INDUSTRY
1926-1940
Reported by Bureau of Labor
tics Plotted on Scale on Left
between 1937 Payrolls and 1937
ction Becomes Basis for
ction Scale on Right
OR MONTHLY PRODUCTION ESTIMATE
OF DOLLARS BY COMPARING "PRO-
TROLL) CURVE WITH SCALE ON RIGHT
ow Production of the Machine
y as Reported in Biennial Census
res
Machinery Division
Foreign and Domestic Commerce
ANNUAL Production Rate in Millions of Dollars
MONTHLY Production Rate in Millions of Dollars
800
700
600
500
400
300
200
100
65
60
55
50
40
35
30
25
20
15
10
5
PRODUCTION as indicated by payrolls
ANNUAL
MONTHLY
EXPORTS
1932
1933
1934
1935
1936
1937
1938
1939
1940
1941
1942
1943
1944
1945

FIGURE 1.

the nice little flower of American industrial progressiveness which hitherto had been the envy and admiration of the whole of the industrial world, with one exception, *i.e.*, Germany.

Can it be wondered, that when National Defense demanded a supreme effort, manufacturers, while recognizing in National Defense the key to their continued existence, neverthelss pointd out that their energetic, enthuiastic participation in the defense effort could only be obtained by the removal of the unpleasant blanket referred to in the preceding paragraph. It will be recollected that this blanket was lifted with a whoop when the five-year accelerated amortization plan was announced as available to all those who might seek salvation by a rush to get under the wire by dead line date.

When a similar scheme for the modernizing of American mechanical industry was discussed with manufacturers previously it was rejected because it was believed (we being at peace) that it contravened a basic principle of democracy, *i.e.*: it rewarded taxation and automatically inversely gave sanction to punitive taxation which was something the conservatives were unwilling to concede to the government in power. The results of N.R.A., plus decreased depreciation allowances plus the decision to permit *laissez faire* policy to govern the rehabilitation of American industry are well known and Figure 1, a graph of business in the United States Machine Tool Industry, speaks eloquently of what transpired. Industry took its own good time to sharpen its tools of production and we entered the present conflict with a banked up demand for machine tools which by comparison made the demands of the last war look like the business of a corner grocery store in a country town of 2000 people.

It was extremely significant that the first thing that the heavy goods manufacturers demanded in the present emergency was an accelerated amortization plan. After much costly delay and debate the mountain produced its mouse in the form of the five-year amortization plan with its accompanying tags of non-reimbursement certificates, etc., etc., etc., ad infinitum. Under the stress of imminent war which could not be seen in 1934-35-36 and '37 the men at the congressional helm offered that which they had previously condemned. The U. S. Government five-year accelerated depreciation plan was the result, after it became apparent that

without it the protective effort of the nation would be fatally delayed.

Current Situation.

It is expected that the machine tool industry will turn out tools to the total value of $750,000,000 this year of 1941 and this is a stupendous amount of money when one considers that the prior peak of 1929 was only $275,000,000. However, this is not the whole story by any means because as will be noted from the price history curve, Figure 2, the unit price of machine tools has about doubled in the interim so that to equal the number of units produced in 1929 at present day values would call for $550,000,000. On the whole therefore, it may be said that the present war effort of the United States Machine Tool Industry is producing 50 per cent more machine tool units than was the case in the previous peak of 1929. As the bank of unfilled orders has increased continuously for each of the last twelve months, it is obvious that the war effort of the U. S. Machine Tool Industry is not yet nearly adequate to accommodate the greatly increased demand for mechanical combatant equipment.

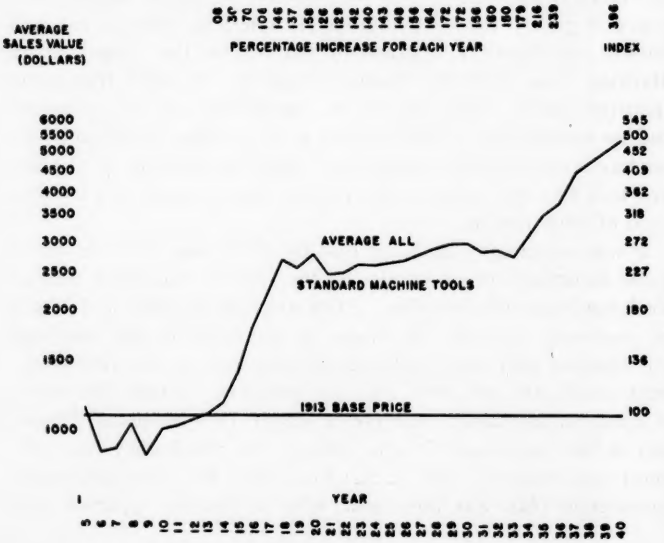


Figure 2.

Concurrently, with any weighing of the debits and credits in the machine tool situation, sight must not be lost of the fact that there has been much technological progress in the effectiveness of all machine tools and on the basis that a price must always be paid for improvement in performance, it is only fair to point out that the productive efficiency of practically all machine tools is about 20 per cent greater than it was ten or twelve years ago.

Machinery has advanced in price partly because of this same better productive capacity of the tools of today, and if a definition of "better" is required it can be said that the tools of today

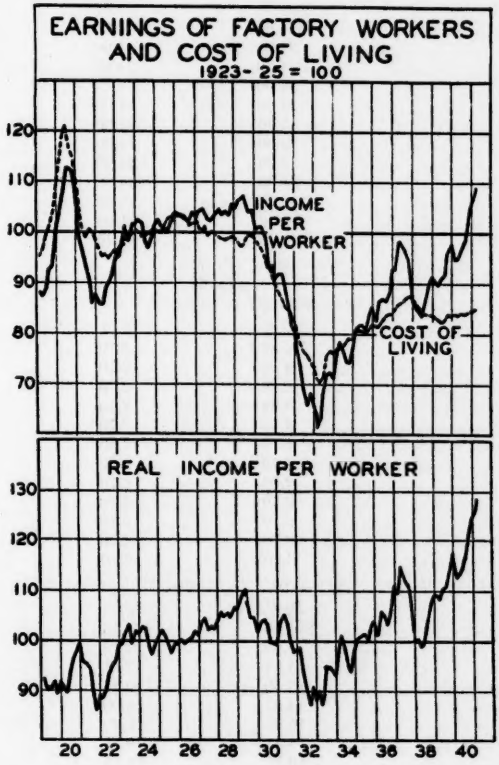


FIGURE 3.

TABLE I.

BASIC MATERIALS—COSTS

PRICES OF PIG IRON, OPEN-HEARTH AND BESSEMER BILLETS, ALLOY BARS (HOT), AND HEAVY MELTING STEEL SCRAP—JANUARY, 1939, THROUGH MAY, 1940 (PER GROSS TON)

Pig Iron, Basic, Valley

	1939	*1940*	*1941*
January	$20.50	$22.50	$23.50
February	20.50	22.50	23.50
March	20.50	22.50	23.50
April	20.50	22.50	23.50
May	20.50	22.50	23.50
June	20.50	22.50	—
July	20.50	22.50	—
August	20.50	22.50	—
September	21.50	22.50	—
October	22.50	22.50	—
November	22.50	22.50	—
December	22.50	23.00	—

Open-Hearth and Bessemer Billets, f.o.b. Pittsburgh

	1939	*1940*	*1941*
January	$34.00	$34.00	$34.00
February	34.00	34.00	34.00
March	34.00	34.00	34.00
April	34.00	34.00	34.00
May	34.00	34.00	34.00
June	34.00	34.00	—
July	34.00	34.00	—
August	34.00	34.00	—
September	34.00	34.00	—
October	34.00	34.00	—
November	34.00	34.00	—
December	34.00	34.00	—

Alloy Bars (Hot)
(cents per pound)
(base, 20 tons or over)

	1939	*1940*	*1941*
January	2.80	2.70	2.70
February	2.80	2.70	2.70
March	2.80	2.70	2.70
April	2.80	2.70	2.70
May	2.75	2.70	2.70
June	2.70	2.70	—
July	2.70	2.70	—
August	2.70	2.70	—
September	2.70	2.70	—
October	2.70	2.70	—
November	2.70	2.70	—
December	2.70	2.70	—

Heavy Melting Steel Scrap, Delivered at Pittsburgh

	1939	*1940*	*1941*
January	$15.60	$18.15	$22.15
February	15.65	17.75	20.75
March	15.75	17.05	20.75
April	15.50	16.45	20.20
May	14.55	18.00	20.00
June	15.00	19.90	—
July	15.55	19.55	—
August	16.15	18.75	—
September	18.75	20.15	—
October	23.15	21.30	—
November	21.85	21.50	—
December	18.50	22.75	—

Source: STEEL.

TAX COLLECTIONS AND NATI
FISCAL YEARS 1913 AND

Year	National Income in Billions of Dollars[1]	Tax Millions of Dollars	Tax Per Cent All Tax Collections
1913	35.4	2,182	6.2
1922	55.6	7,502	13.5
1923	64.6	7,234	11.2
1924	69.3	7,812	11.3
1925	71.7	7,884	11.0
1926	75.3	8,605	11.4
1927	75.8	9,059	12.0
1928	77.2	9,342	12.1
1929	80.9	9,759	12.1
1930	75.9	10,266	13.5
1931	61.7	9,300	15.1
1932	47.2	8,146	17.3
1933	41.3	7,500	18.2
1934	46.6	8,771	18.8
1935	53.7	9,730	18.1
1936	61.0	10,505	17.2
1936[a]	61.0	10,472	17.2
1937	68.6	12,503	18.2
1937[a]	68.6	11,902	17.3
1938	68.0	14,072	20.7
1938[a]	68.0	12,651	18.6
1939	66.3[b]	13,677	20.6
1939[a]	66.3[b]	12,131	18.3
1940	71.8[b]	14,341	20.0
1940[a]	71.8[b]	12,648	17.6

1. Income produced estimates by Department
1939 and 1940; figures are averages of t
for the year indicated and the preceding ye
1939 and 1940 are the sums of income
months ended June 30 of each year.
a. Exclusive of employment or pay roll taxes.
b. National income payments for year ende

TAXES GE

Taxes Related to P
Taxes Related to Vo
or Expense
Taxes Levied on Pr

* Excluding manufac
turers of tobacco.

TABLE II.

TAXES.

AND NATIONAL INCOME S 1913 AND 1922-1940

Tax Collections Per Cent of National Income

ns rs	All Tax Collections	Federal	State	Local
2	6.2	1.9	0.8	3.4
2	13.5	6.3	1.5	5.7
4	11.2	4.7	1.4	5.1
2	11.3	4.6	1.5	5.2
4	11.0	4.1	1.5	5.3
5	11.4	4.3	1.7	5.5
9	12.0	4.4	1.8	5.8
2	12.1	4.1	2.0	6.0
9	12.1	4.1	2.0	6.0
6	13.5	4.6	2.3	6.6
0	15.1	4.4	2.9	7.8
6	17.3	3.8	3.5	10.0
0	18.2	4.3	3.6	10.2
1	18.8	6.2	3.7	8.9
0	18.1	6.6	3.5	8.0
5	17.2	6.3	3.9	7.0
2	17.2	6.3	3.8	7.0
8	18.2	7.3	4.5	6.4
2	17.3	7.0	4.0	6.4
2	20.7	8.7	5.3	6.7
1	18.6	7.7	4.3	6.7
7	20.6	8.2	5.4	7.0
1	18.3	7.0	4.2	7.0
1	20.0	7.8	5.4	6.8
8	17.6	6.6	4.2	6.8

y Department of Commerce except for averages of the calendar year estimates preceding year. The amounts used for e of income payments for the twelve h year.
ay roll taxes.
r year ended June 30.

TAX COLLECTIONS IN THE UNITED STATES

FISCAL YEARS 1913 AND 1922-1940

Millions of Dollars

Year	Total	Federal[1]	State	Local
1913	2,187	668	300	1,219
1922	7,502	3,487	858	3,157
1923	7,234	3,032	917	3,285
1924	7,812	3,193	1,017	3,602
1925	7,884	2,966	1,107	3,811
1926	8,605	3,207	1,264	4,134
1927	9,059	3,337	1,355	4,367
1928	9,342	3,194	1,507	4,641
1929	9,759	3,328	1,612	4,819
1930	10,266	3,468	1,780	5,018
1931	9,300	2,717	1,778	4,805
1932	8,146	1,788	1,642	4,716
1933	7,500	1,785	1,505	4,210
1934	8,771	2,890	1,721	4,160
1935	9,730	3,545	1,886	4,299
1936	10,505	3,845	2,370	4,290
1936[a]	10,472	3,845	2,338	4,289
1937	12,503	5,028	3,105	4,370
1937[a]	11,902	4,775	2,759	4,368
1938	14,072	5,934	3,595	4,543
1938[a]	12,651	5,220	2,893	4,538
1939	13,677	5,412	3,609[b]	4,656
1939[a]	12,131	4,672	2,810[b]	4,649
1940	14,341	5,566	3,875	4,900
1940[a]	12,648	4,727	3,028	4,893

1. Federal tax collections less refunds.
a. Exclusive of employment or payroll taxes.
b. An estimated $275 million of local shares were deducted from state tax collections for 1939.

Source: United States Treasury Department, Bureau of the Census, National Industrial Conference Board; state taxes for 1938 and local taxes for 1938, 1939 and 1940 are preliminary estimates by the Tax Foundation.

TAXES GROUPED ACCORDING TO MANAGEMENT VIEWPOINT

(as percentages of total taxes paid)

	*Manufacturing** *Per Cent*	*Wholesaling** *Per Cent*	*Retailing* *Per Cent*
elated to Permanent Assets	24	28	27
elated to Volume of Sales ense	43	38	54
evied on Profit	33	34	19
	100	100	100

ling manufacturers and wholesalers of alcoholic beverages and petroleum products and manufac- obacco.

REVIEW OF MACHINE T

Machine	*Type*	*1938*
Drills, Radial	9″ col., 36″ arm	$2,346.00
with motor and	17″ col., 72″ arm	$8,839.00
coolant system	19″ col., 96″ arm	$10,559.00
Planer, complete	36″ × 36″ × 10′ Openside Hyd.	$15,337.00
with motor and	Two rail heads, one	
control	side head	
Shaper, complete	24″ hydraulic Universal	$3,871.00
with motor, control		
and power down feed		
Grinder, complete	#2-10″ × 27″ Universal	$1,716.00
with motor and	Tool and Cutter	
control		
Milling Machine	#2-L-Univ. Light Type	$3,767.00
with motor and	#2-HS-Universal	$6,108.00
coolant system		
Lathe, Turret	#3 Univ. Ram Type	$3,505.00
with motor, control	1-1/2″ bar capacity	
and coolant system		
Horizontal Boring,	330T Table Type	$12,470.00
Drilling, Milling		
Machine with Motor,		
Control and Demonstrator		

TABLE III.

ACHINE TOOLS—PRICES AND DELIVERIES

Average Price			Plus Per Cent		Average Delivery Days			Plus Days	
1938	*1940*	*1941*	*1940*	*1941*	*1938*	*1940*	*1941*	*1940*	*1941*
$2,346.00	$2,927.00	$3,314.00	24	41	66	160	270	94	204
$8,839.00	$10,450.00	$11,377.00	18	29	60	310	320	250	260
$10,559.00	$12,813.00	$13,804.00	21	31	60	352	360	292	300
$15,337.00	$16,942.00	$18,657.00	10	22	120	150	225	30	135
$3,871.00	$4,202.00	$4,788.00	9	23	120	210	270	90	150
$1,716.00	$1,871.00	$2,065.00	9	20	30	120	270	90	240
$3,767.00	$4,342.00	$4,727.00	15	25	57	205	250	148	193
$6,108.00	$7,049.00	$7,440.00	15	22	70	200	260	130	190
$3,505.00	$4,124.00	$4,367.00	17	24	120	206	332	86	212
$12,470.00	$14,199.00	$15,141.00	14	21	130	290	345	160	215

accomplish more of a process in less time, with less raw material, with less power per unit of work performed, than was formerly the case.

Economic Position.

Apart from technological advances, machine tools have advanced in price for three definite reasons: the cost of labor has increased during the period under review as indicated in Figure 3; the cost of materials has increased in the same period as indicated in Table I; and machine tool builders' tax payments have increased as shown in Table II.

A typical cross section of the result of these advances in the three major elements making up machine tool costs is revealed in the tabulation Table III. (Price comparison years 1939-40.)

Labor.

The comparative freedom from labor difficulty enjoyed by the machine tool industry has been achieved by the two factors revealed in Figure 3 and Table III. It should be borne in mind that this is not by any means the whole picture, as investigations of the Bureau of Labor Statistics show that 75 per cent of all workers in the machine tool industry were working on one shift at the end of 1940. What actually happened was that a very large proportion of these men were working a considerable amount of overtime and their total earnings were augmented proportionately. Labor conditions in the Machine Tool industry are as described in Figures 4, 5, and 6. Current thought on the labor situation tends in some respects to be erroneous in concept, for instance one authority indicates an exhaustion of available labor supplies in the near future. As this commentator has in mind skilled machine tool mechanics his conclusion is wrong because the available supply was exhausted a year ago and the new men fed into the industry in the interim have been trained by the industry and the trade schools. Future reinforcements without doubt will have to come from these same two sources.

Sub-contracting is not the answer to the accelerated machine tool production problem because to build good and accurate machine tools requires a degree of supervision and integrity of accuracy

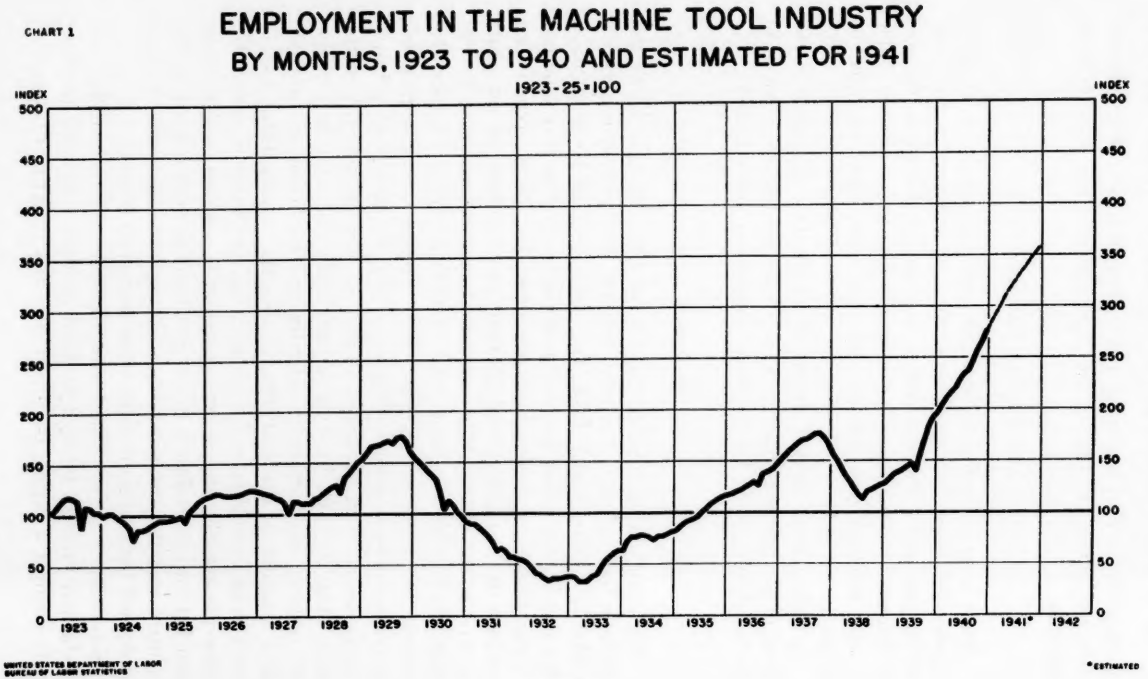


CHART 1
EMPLOYMENT IN THE MACHINE TOOL INDUSTRY
BY MONTHS, 1923 TO 1940 AND ESTIMATED FOR 1941
1923-25=100
INDEX
500
450
400
350
300
250
200
150
100
50
0
1923
1924
1925
1926
1927
1928
1929
1930
1931
1932
1933
1934
1935
1936
1937
1938
1939
1940
1941*
1942
UNITED STATES DEPARTMENT OF LABOR
BUREAU OF LABOR STATISTICS
*ESTIMATED

FIGURE 4.

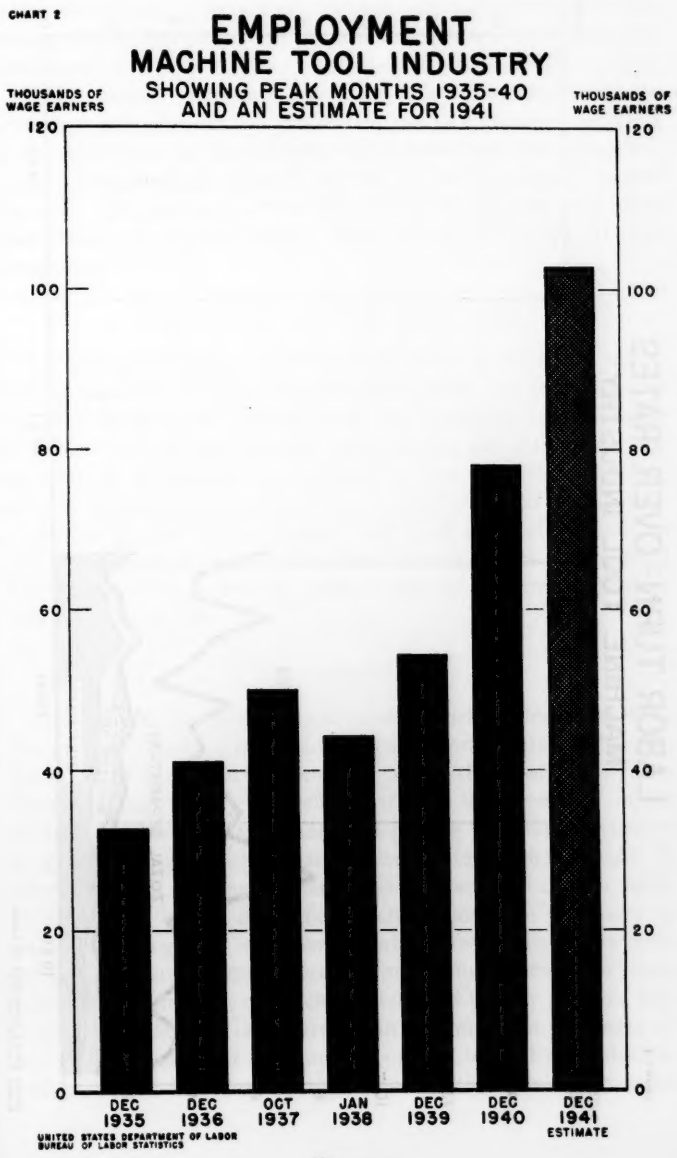


CHART 2
EMPLOYMENT
MACHINE TOOL INDUSTRY
SHOWING PEAK MONTHS 1935-40
AND AN ESTIMATE FOR 1941
THOUSANDS OF WAGE EARNERS
THOUSANDS OF WAGE EARNERS
120
100
80
60
40
20
0
DEC 1935
DEC 1936
OCT 1937
JAN 1938
DEC 1939
DEC 1940
DEC 1941 ESTIMATE
UNITED STATES DEPARTMENT OF LABOR
BUREAU OF LABOR STATISTICS

FIGURE 5.

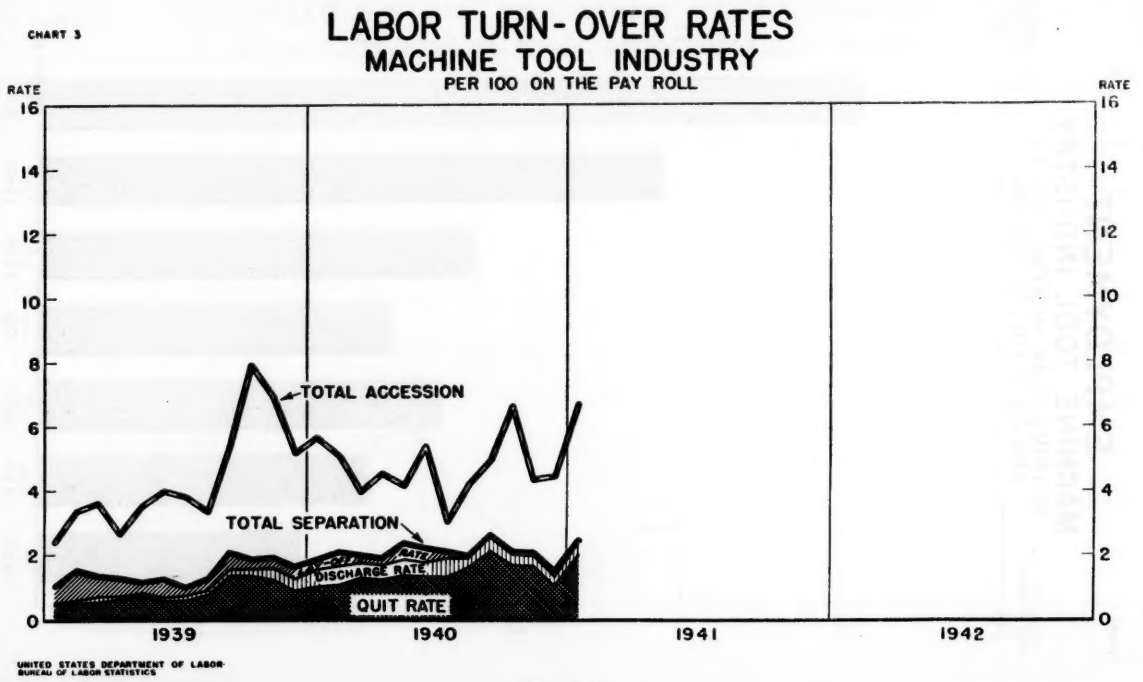


CHART 3
LABOR TURN-OVER RATES
MACHINE TOOL INDUSTRY
PER 100 ON THE PAY ROLL
RATE
16
14
12
10
8
6
4
2
0
TOTAL ACCESSION
TOTAL SEPARATION
DISCHARGE RATE
QUIT RATE
1939
1940
1941
1942
UNITED STATES DEPARTMENT OF LABOR
BUREAU OF LABOR STATISTICS

FIGURE 6.

which the average sub-contractor cannot supply for the very simple reason that he has never been trained to either see or produce this accuracy.

While greatly increased sub-contracting is undoubtedly the answer to many production problems it can be eliminated definitely as a major factor in the machine tool industry and the sooner this fact is recognized the better it will be for our all around national security. An analysis of machine tool building costs will show that there is in normal times a very substantial amount of sub-contracting.

Conservation of Productive Time.

Far-seeing machinery users have for years realized that lathes, milling machines, drilling machines, and planers use only about 26 per cent of the total available time for actual cutting operations and this characteristic indicates wherein lies the major field for speeding up production and effecting economies. Better tooling service, permitting more continuous cutting of metal to be maintained, is worthy of the closest study and it should be set up as a national objective that all machine tools should be so employed that they are actually cutting metal at least 50 per cent of the time they are in operation.

Management.

In all production problems the most important element making for success of lack of it is management. Fortunately, we have in the ranks of the American working people the most industrious, ingenious, and "highly appreciative of good living people" in the world. Furthermore, these same people are the most responsive to good and progressive management. Hence the thought is offered without any serious danger of refutation, that such blow-holes as exist in our economic structure today are due more to the shortcomings of management which should have exercised the skill and farsightedness necessary to eliminate them. To blame our economic distress, our lack of ability to fill our machine tool orders, the delays in our rearmament program, the shortages of anti-aircraft guns and ammunition, on the lack of an unlimited well of skilled labor in the machine tool field is to take a stand

FIGURE 6.

which merits what is technically known as a good "horse laugh" providing, of course, one has the strength and sense of humor to extract a laugh out of a situation that tends to become too tragic to be funny.

Conclusion.

No less an authority than T.N.E.C. has stated that "Democracy will fail if we destroy profit economy" and no quarrel can be had with this basic thought. With increasing labor and material costs, selling prices must advance if we are to preserve this profit economy.

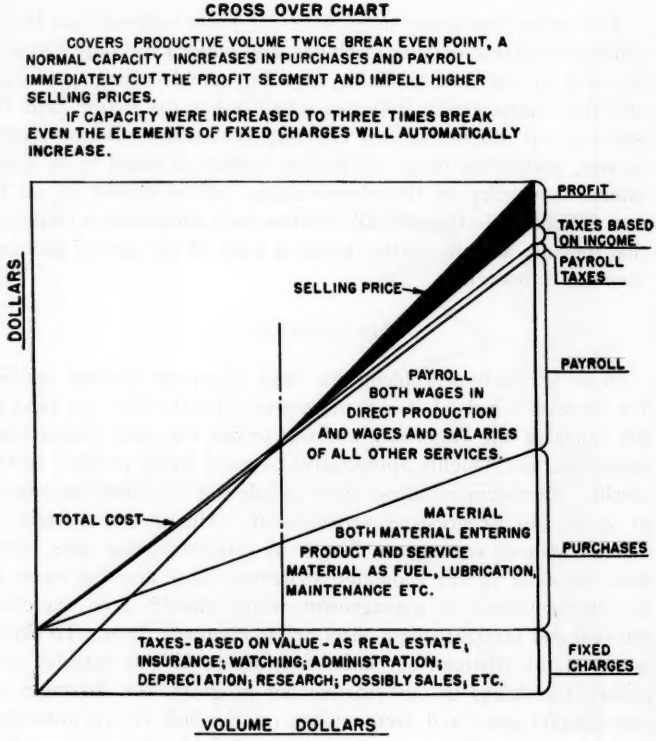

CROSS OVER CHART
COVERS PRODUCTIVE VOLUME TWICE BREAK EVEN POINT, A NORMAL CAPACITY INCREASES IN PURCHASES AND PAYROLL IMMEDIATELY CUT THE PROFIT SEGMENT AND IMPELL HIGHER SELLING PRICES.
IF CAPACITY WERE INCREASED TO THREE TIMES BREAK EVEN THE ELEMENTS OF FIXED CHARGES WILL AUTOMATICALLY INCREASE.
DOLLARS
PROFIT
TAXES BASED ON INCOME
PAYROLL TAXES
SELLING PRICE
PAYROLL
PAYROLL BOTH WAGES IN DIRECT PRODUCTION AND WAGES AND SALARIES OF ALL OTHER SERVICES.
TOTAL COST
MATERIAL BOTH MATERIAL ENTERING PRODUCT AND SERVICE MATERIAL AS FUEL, LUBRICATION, MAINTENANCE ETC.
PURCHASES
TAXES - BASED ON VALUE - AS REAL ESTATE; INSURANCE; WATCHING; ADMINISTRATION; DEPRECIATION; RESEARCH; POSSIBLY SALES; ETC.
FIXED CHARGES
VOLUME DOLLARS

Figure 7.

Furthermore, prices will naturally advance in reflection not only of increases in the costs of labor and material directly employed in the production of a given piece of machinery but also in proportionate measure to increasing costs of supplies and the materials as well as the casual and unskilled labor which services productive labor. Figure 7 illustrates the variable and non-variable elements in manufacturing costs in the way in which they are applied in a typical machinery building company's accounting.

The ultimate in our thinking on this subject therefore gradually becomes obvious, *i.e.*: even in wartime, political, industrial and social leadership should be devoted to the production of an increased quantity of better quality goods, whether these goods be swords or plowshares. It is recognized that this objective can only be obtained by improved mechanical processes which in themselves cost a progressively increasing amount of money but which are more than offset by a lowering of ultimate costs and the spreading of the benefits of mechanization to all the people; in the case of industrial goods—by a better standard of living, and in the case of war goods—by victory for our side, without which in complete form, nothing would have value, relative or direct.

DISCUSSIONS OF "STOPPING AND BACKING TRIALS OF A DESTROYER."

By Lt. Commander P. D. Gold, Jr., U.S.N.
Volume 53, No. 1, February, 1941.

By Lt. Commander John M. Labberton, U.S.N.R. *

Commander Gold's paper is of a great deal more importance and interest than might be first suspected. These tests indicate certain fundamental phenomena which, when put in basic form, may enable us to compute with great accuracy the performance of any propeller in stopping a ship when a known torque is exerted in reverse.

Captain Smith, in his discussion, used a curve of propeller revolutions versus ship's speed in knots. (His Figure 2). This "actual ship curve" was taken from the data on Commander Gold's Figure 11. However, it should be noted that Captain Smith's Figure 2 holds not only for the data on Figure 11 *but for all the runs shown from Figure 11 to Figure 24 inclusive.* This is because the same torque is available for any given RPM of the turbines with full astern throttle steam conditions.

In other words, with the ship moving at a certain speed, the speed-torque curve of the turbines being fixed for this particular ship and throttle condition, the turbine or propeller RPM will naturally take or assume a certain fixed value. All of the tests clearly show this.

The stronger or greater the torque of the turbine, the more nearly will this curve approach the A B C line in Captain Smith's Figure 2.

The speed-torque curve mentioned above must have subtracted from it, when considering its effect on ship deceleration, that torque required to decelerate the rotating mass system.

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* Associate Professor of Mechanical Engineering, New York University.

To consider the situation from a fundamental standpoint we can start with the expression

$$T = \rho D^2 v^2 \left[f\left(\frac{nD}{v}\right)\right]$$

where

T = Propeller thrust in pounds

D = Propeller diameter in feet

n = Propeller revolutions per second

v = Propeller speed of advance in feet per second

ρ = Density of fluid in pounds per cubic foot

This can be proven by means of dimensional analysis for the case of any propeller working in a viscous fluid and holds so long as the propeller is not in cavitation.

From this expression we can obtain several dimensionless coefficients which will be of use to us in further analyzing Commander Gold's tests.

If

$$T = \rho D^2 v^2 \left[f\left(\frac{nD}{v}\right)\right]$$

then

$$\rho \frac{T}{D^2 v^2} = f\left(\frac{nD}{v}\right)$$

The left hand member is a dimensionless coefficient based on slip. However, we can discard ρ because we will always use the same fluid and we can change the other terms to nautical units and obtain a quasi-dimensionless coefficient which is fully as useful to us and more convenient to handle. In short, change n to N for RPM and v to V for knots. Then we have

$\frac{T}{D^2V^2}$ which remains fixed for a given value of

$\frac{ND}{V}$ the slip function.

In the same manner, for a given pitch, since any part of the blade face is an inclined plane acting at a certain distance from the center of rotation, we can change the thrust coefficient to a

torque coefficient by multiplying both numerator and denominator by a linear dimension proportional to diameter or

$$\frac{T}{D^2V^2} = \frac{TDk}{D^2DV^2k} = \frac{TDk}{D^3V^2k} = \frac{Q}{D^3V^2k}$$

Where Q = pound-feet torque.

Eliminating k as a constant we obtain

$$\frac{Q}{D^3V^2}$$ as another quasi-dimensionless coefficient

which remains fixed for a given value of

$$\frac{ND}{V}$$, the slip function.

The values of these coefficients at various speeds will now be determined for the series of tests made with full astern throttle. See Table I.

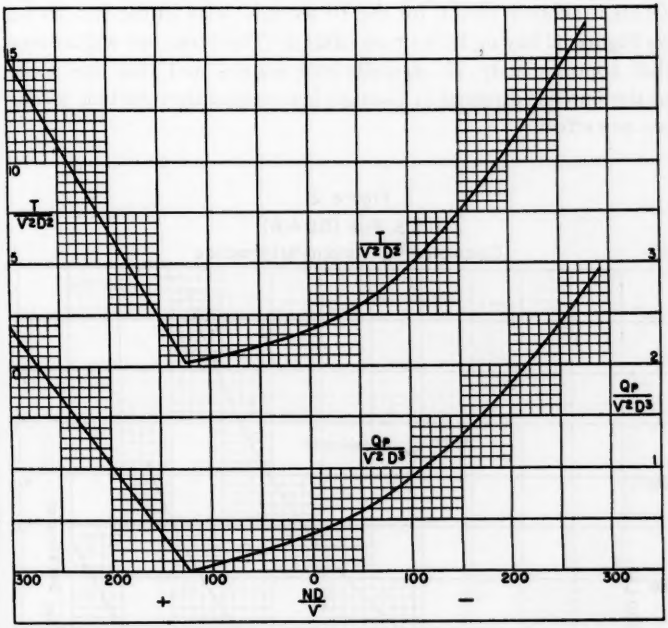
Not knowing the value of the wake fraction for this ship, the value of the actual ship speed rather than the speed of advance of the propeller is used in determining the values for the coefficients in the table. Since the resultant curve will be used to calculate performance of the same ship, this procedure is permissible. However, if this propeller performance curve were to be used in predicting its performance with another ship, correction for wake factor would be necessary.

In ships of this type, wake fraction remains approximately constant over the complete range of speed. It therefore appears to the writer that if the ship is decelerating the wake speed should continually bear the same constant relation to the ship speed, the ship *always* moving faster than the wake and running away from it. Even at zero speed this must hold true, otherwise the ship would be floating in its moving wake and not be dead in the water. Wake fraction, therefore, is not corrected for, as we will go into and out of the curves with the same wake and same ship.

For the various values of speed in knots, the values of N and T were taken from Commander Gold's Figure 36 and Captain Smith's Figure 2, respectively. The values of torque were taken from all of the runs, Figures 11 to 24, inclusive, and it will be noted that as regards speed and torque, these curves all check each other very well.

From Table I, the set of curves on Figure 1 were drawn.

Figure I
U.S.S. Roe (DD 418)
Propeller Performance Reverse Torque

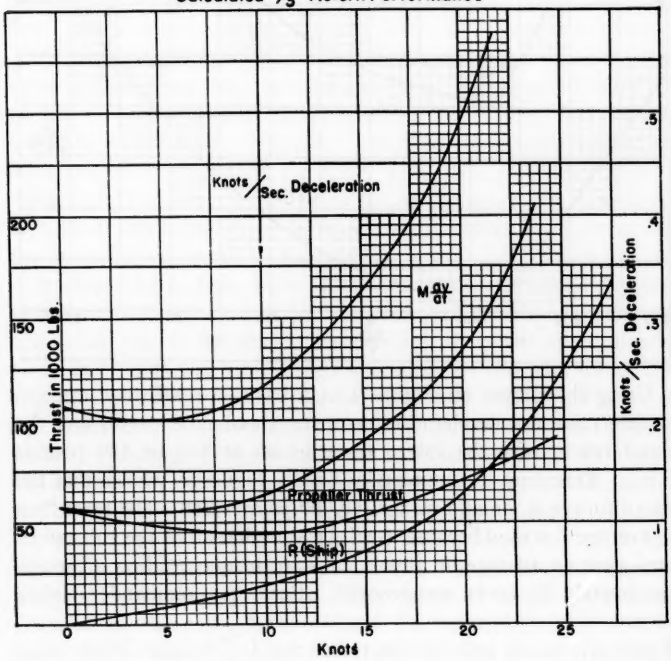


Using the curves on Figure 1 as a basis, we will now attempt to determine the deceleration rate, the speed-time curve, and the ahead reach, with the astern throttles set at 2/3 or 150 pounds P.S.I. Referring to Commander Gold's Figure 2, we see that the astern torque is, on an average, 60,000 pound-feet. The term " on an average " is used because it will be seen that all of the full astern runs gave an average or almost constant torque reading after approximately 23 knots was reached. Refer to Commander Gold's Figure 37 and it will be noted that the $I \frac{d\omega}{dt}$ torque value was,

on an average, 10,000 pound-feet. This, of course, must be absorbed by the astern turbine, leaving an effective torque on the propeller of approximately 60,000-10,000 = 50,000 pound-feet.

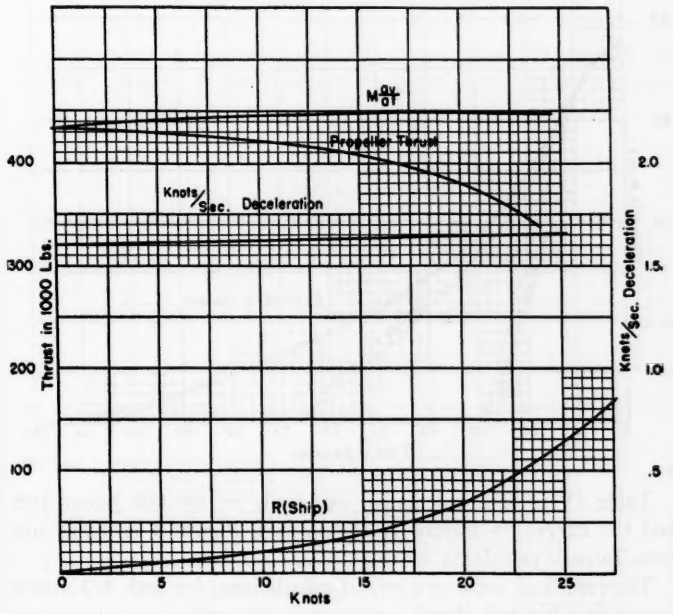
Reference to Commander Gold's Figure 11 will disclose that the astern turbine did not really come into play until approximately 20 seconds had elapsed due to the high inertia of the rotating system. (This is another reason for the 10 seconds delay in the run shown on Figure 32 having little or no effect.) Therefore, we will assume that approximately 20 seconds time elapses and that the speed of the ship has dropped to 23 knots before the astern turbine torque has any effect.

Figure 2
U.S.S. Roe (DD 418)
Calculated 2/3 Astern Performance



Therefore, starting with 23 knots and 50,000 pound-feet torque, from Figure 1, we can make Table II. From Table II, Figure 2 can be drawn which is similar to Commander Gold's Figure 36. In fact the ship resistance is taken therefrom. This curve, however, is a *calculated* curve for 2/3 astern whereas Commander Gold's Figure 36 is derived from tests.

Figure 3
U.S.S. Roe (DD418)
Calculated Performance Full Ahead Power Astern



From Figure 2, Table III is made. The results of this are plotted over Commander Gold's Figure 35 which is derived from tests. Thus it can be seen that the calculated prediction of the 2/3 astern performance is very close to the tested performance. See Figure 4.

In a similar manner, the performance for any degree of astern torque can be predicted. For instance, let us assume that the U.S.S. *Roe* is an electric drive ship and that full ahead torque is available astern. This amounts to approximately 400,000 pound-feet per propeller. See Commander Gold's Figure 11.

Figure 4
U.S.S. Roe (DD418)
Deceleration Curves

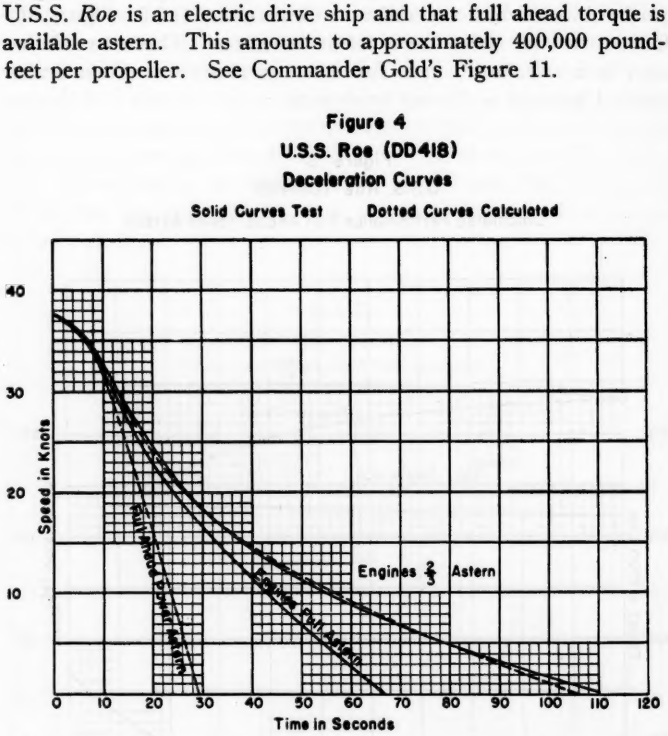


Table IV is first calculated, using $Q_p = 400{,}000$ pound-feet and the curves on Figure 1. From this, Figure 3 is made and then Table V calculated.

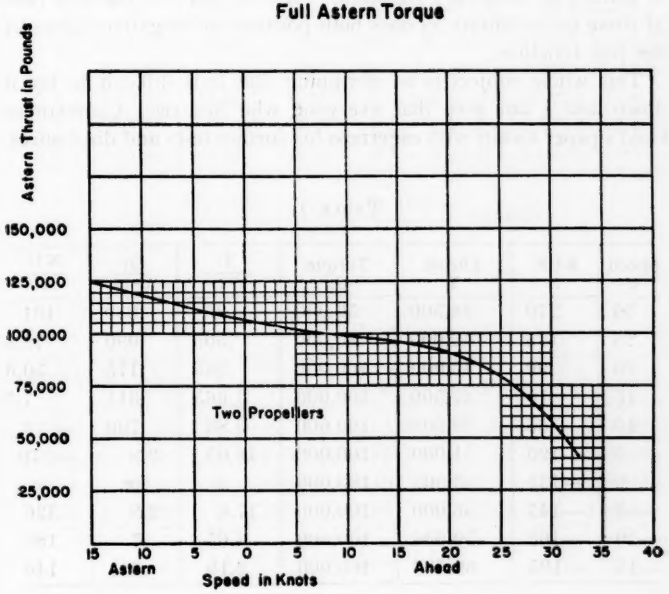
The results of these two sets of calculations, for both 2/3 astern power and for full ahead power operating action, are plotted on Commander Gold's Figure 35 (which is reproduced in this discussion) as Figure 4.

The area under these curves multiplied by the proper constant is, of course, the ahead reach and this value for the full ahead power astern is 370 yards.

Captain Smith has mentioned the bugaboo of the tremendous hump or peak of negative torque in going from full ahead RPM to zero RPM. Naturally, from the foregoing demonstration, it is impossible to get this hump because the astern torque depends on the engines themselves. Commander Gold's Figures 29, 30 and 31 demonstrate this beautifully. Whether or not the propellers were cavitating makes no difference and I am convinced that this test result *can* be applied to other ships, provided proper correction is made for hull and propeller characteristics. I can't see that there is any cause for worry about this possible heavy torque.

Captain Smith expressed the hope, in his discussion, that Commander Gold could repeat these careful experiments with an electric drive battleship. It would certainly be a great contribution to the solution of this problem and should lay for all time the ghost of the "hump" theory. It would be interesting to obtain, in advance, the

Figure 5
U.S.S. Roe (DD418)
Full Astern Torque



characteristic curve of the battleship propeller in the water tunnel. This could be done by running at one water velocity and varying propeller speeds, positive and negative, and recording thrusts and torques. Then, having the E.H.P. curve of the battleship and data as to the inertia of the rotating system the performance of the ship could be predicted and this prediction checked with the test. This, of course, could be done in the case of any ship.

So far no mention has been made of the backing performance—only the stopping. However, on Figure 1 and Figure 5 are shown the backing characteristics. These can be used to calculate backing performance in a manner similar to that in which the stopping performance was calculated. The astern thrust shown on Figure 5 is a mere continuation of the thrust shown on Commander Gold's Figure 36, and was calculated from the data on Figure 11. It will be noted that the quasi-dimensionless coefficients for both thrust and torque touch zero for a certain positive value of the slip function. This is the point for infinite speed of the ship either forward or astern, an imaginary situation. At zero speed of the ship both of these go to infinity as does both positive and negative values of the slip function.

This whole subject is so intriguing that it is difficult to lay it down and I am sure that everyone who has read Commander Gold's paper awaits with eagerness his further tests and disclosures.

TABLE I.

Speed V	RPM N	Thrust T	Torque Q_p	$\frac{T}{V^2D^2}$	$\frac{Q_p}{V^2D^3}$	$\frac{ND}{V}$
30	270	28,500	50,000	.25	.039	101
25	175	40,000	80,000	.505	.090	78.8
20	90	45,000	100,000	.885	.175	50.6
15	10	47,500	100,000	1.665	.311	7.5
10	—65	49,000	100,000	3.86	.700	—73
5	—120	51,000	100,000	16.05	2.8	—270
0	—135	53,000	100,000	∞	∞	∞
—5	—145	56,000	100,000	17.6	2.8	326
—10	—165	59,000	100,000	4.65	.7	186
—15	—195	62,500	100,000	2.19	.311	146

TABLE II.

Speed V	$\frac{Q_p}{V^2D^3}$	$\frac{ND}{V}$	$\frac{RPM}{N}$	$\frac{T}{V^2D^2}$	T
23	.066	80	163	.63	42,100
15	.155	55	73.5	.9	25,700
10	.35	0	0	1.7	21,600
5	1.4	—150	—66.5	7.5	23,800
3	3.9	—350	—93.5	25	28,600

$Q_p = 50{,}000$ pound-feet

TABLE III.

Speed V	$M\frac{dv}{dt}$	$\frac{dv}{dt}$ ft./sec.	Knots/ Sec.	Speed Change Knots	Seconds Required
21.5	155,000	.963	.57	23 — 20 = 3	3/.57 = 5.25
17.5	106,000	.66	.39	20 — 15 = 5	5/.39 = 12.8
12.5	75,000	.466	.276	15— 10 = 5	5/.276 = 18.1
7.5	56,000	.348	.206	10 — 5 = 5	5/.206 = 24
2.5	55,000	.342	.202	5 — 0 = 5	5/.202 = 25

$$M = \frac{2315 \text{ tons} \times 2240 \text{ lbs/ton}}{32.2} = 161{,}000$$

TABLE IV.

Speed V	$\frac{Q_p}{V^2D^3}$	$\frac{ND}{V}$	$\frac{RPM}{N}$	$\frac{T}{V^2D^2}$	T
23	.53	—40	—81.6	2.6	175,000
15	1.24	—145	—193.5	7.2	206,000
10	2.8	—280	—249	17	216,000
—10	2.8	325	—289	16.5	210,000

$Q_p = 400{,}000$ pound-feet

TABLE V.

Speed V	$M\frac{dv}{dt}$	$\frac{dv}{dt}$	Knots/ Sec.	Speed Change Knots	Seconds Required
				32 — 23 = 9	9/1.65 = 5.45
21.5	448,000	2.78	1.645	23 — 20 = 3	3/1.645 = 1.82
17.5	448,000	2.78	1.645	20 — 15 = 5	5/1.645 = 3.04
12.5	446,000	2.77	1.64	15 — 10 = 5	5/1.64 = 3.05
7.5	442,000	2.74	1.62	10 — 5 = 5	5/1.62 = 3.09
2.5	438,000	2.72	1.61	5 — 0 = 5	5/1.61 = 3.1

By D. C. MacMillan,* Associate Member.

Commander Gold is to be congratulated on the excellence of his paper, and the data presented therein. We are indebted to the Navy Department for publication of the results.

The description of the instrumentation and procedure employed greatly increases the value of the paper. The substitution of cameras for human observers and recorders renders the data more accurate and reliable. The use of shore stations, directed by radio, to determine the instantaneous ship position during deceleration is much more reliable than range finders on shipboard, which are often rendered nearly useless by vibration set up during the cavitation period.

During the past three years, the writer has attempted to collect and assimilate data on the stopping trials of about twenty merchant vessels of four types. The instrumentation employed was not as adequate as Commander Gold's, and the results in some cases were worthless. However, some reliable data was secured, which is presented herein, and may be of interest in amplifying the existing information on the subject.

The hull and propeller characteristics are given in Table 1. It will be noted that the five ships represent three classes. The trials of the first three vessels were conducted at Rockland, Maine, while those of the last two were run off New York, either near Ambrose or Fire Island Light ships.

The apparatus and procedure used is briefly described below. The instantaneous ship position was determined by use of stadimeter or range finder and compass bearings. These instruments were furnished and manned by members of the United States Maritime Commission Trial Board, and the results given herein were taken from the trial reports of the various ships.

* Federal Shipbuilding & Dry Dock Company.
(Statements and interpretations of data in this discussion are to be understood as individual expressions of the author and not those of the Federal Shipbuilding & Dry Dock Company)

TABLE I. SHIP AND PROPELLER CHARACTERISTICS.

Ship	*Markay*	*Neosho*	*Red Jacket*	*Sea Fox*	*Fred. Lykes*
HULL					
Designed					
Length, W.L. ft.	543	543	438	469.5	469.5
Beam, ft.	75.0	75.0	63.0	69.5	69.5
Draft, moulded, ft.	29.88	29.88	25.75	27.25	27.25
Displacement, tons	23,200	23,200	13,812	16,725	16,725
Trial					
Draft, ft.	29.91	30.02	26.00	18.5	18.48
Trim by stern, ft.	0	0.05	0.16	10.5	10.54
Displacement, tons	23,104	23,275	13,910	10,577	10,389
PROPELLERS					
Number	2	2	1	1	1
Design SHP, each	6750	6750	6000	8500	8500
Design RPM	96	96	92	85	85
Number blades	4	4	4	4	4
Diameter, ft.	17.5	17.5	19.0	21.67	21.67
Pitch, ft. (a)	22.19	22.19	19.875	21.67	21.67
Dev. area, sq. ft.	111.6	111.6	132.7	166.4	166.4
Blade thick. fraction	.0477	.0477	.0435	.044	.044
Tip immersion, ft.	9.2	9.3	5.5	0.9	0.95
STOPPING DATA					
Time to stop, seconds					
Shaft	44	32	25.6	29.0	22.0
Ship	353	285	300	204	162
Head reach, yards	1415	1265	1680	1040	895

(a) Pitch at 0.7 radius

An experimental installation of a pitot tube to measure instantaneous speed was made by the writer on the *Markay* and *Neosho*. The tube was calibrated during the standardization trials of the vessels, since no model calibration was available.

The signal circuit consisted of a bell and light at each observing station, and was operated by an observer on the bridge. Three and two bell warnings were given before the start of each run, and one bell at the start and every 30 seconds thereafter. Torsion-

meters, revolution counters, etc., were read at more frequent intervals, the time being determined by stop-watches synchronized with the starting signal.

One Denny-Edgecombe mechanical optical torsionmeter was fitted on each shaft. Three observers were used for each meter; one to read torque, the second to read the stop-watch, and the third to record both readings. This procedure was used to obviate the difficulty of reading this type of meter at a given time or signal when the dial might not be visible or the reading steady. Readings were taken as often as possible for two or three minutes, and every 30 seconds thereafter.

A Kingsbury thrust meter was installed on the *Red Jacket*. A procedure similar to the above was used to record data.

Shaft revolutions were determined by counter and tachometer. A Smith-Cummings counter was installed on each shaft, and was shifted at 10-second intervals by the bridge observer. Another observer recorded the readings. The tachometer was observed at 5-second intervals, one man giving the signal (based on stop-watch time) and recording, another reading the instrument. The above observations were made at the 5 or 10-second intervals for two minutes, and every 30 seconds thereafter.

The trial results are given herein, as follows:

Figures 1 to 3. Ship speed on time.

Figures 4 to 8. Torque and ship speed on revolutions.

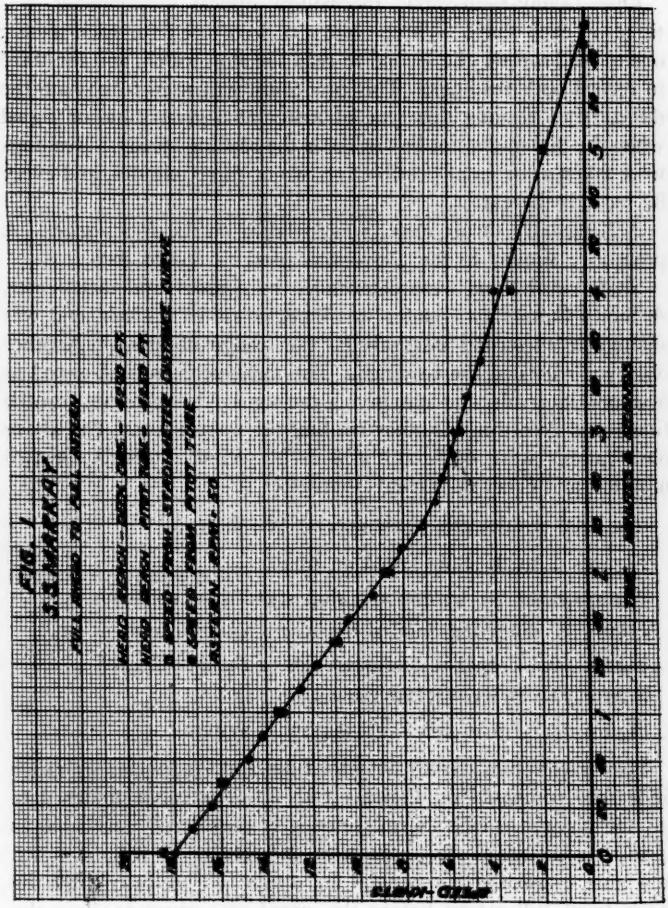
Figure 9. Stopping data of the *Red Jacket* on time.

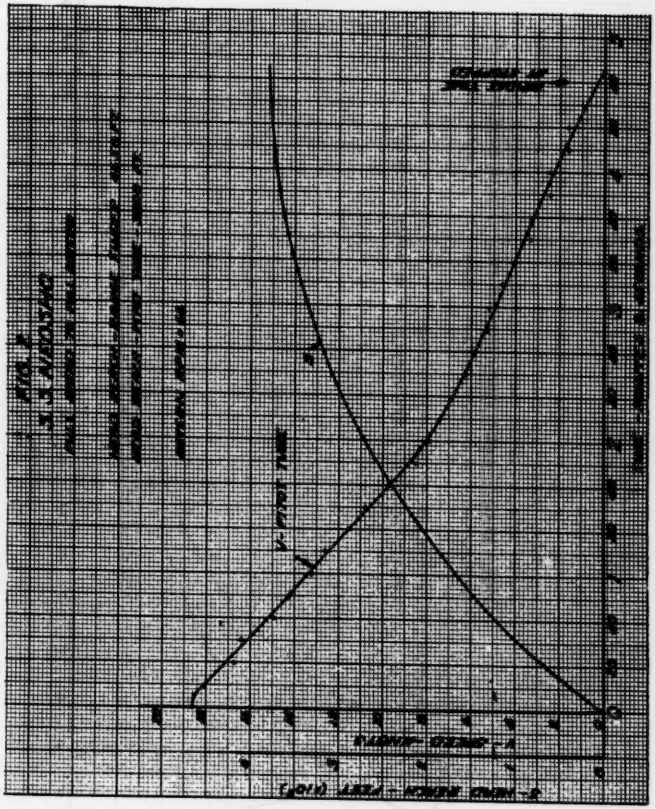
Figure 10. Propeller characteristics of the *Red Jacket*.

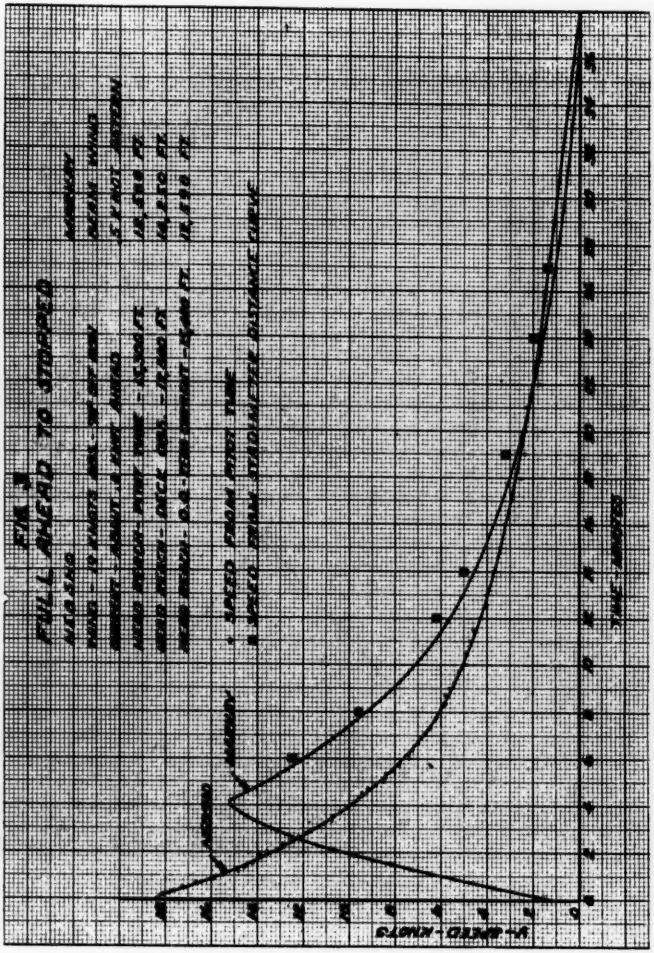
Figure 11. Speed—RPM curves.

Figure 12. Force curves of ship mass system.

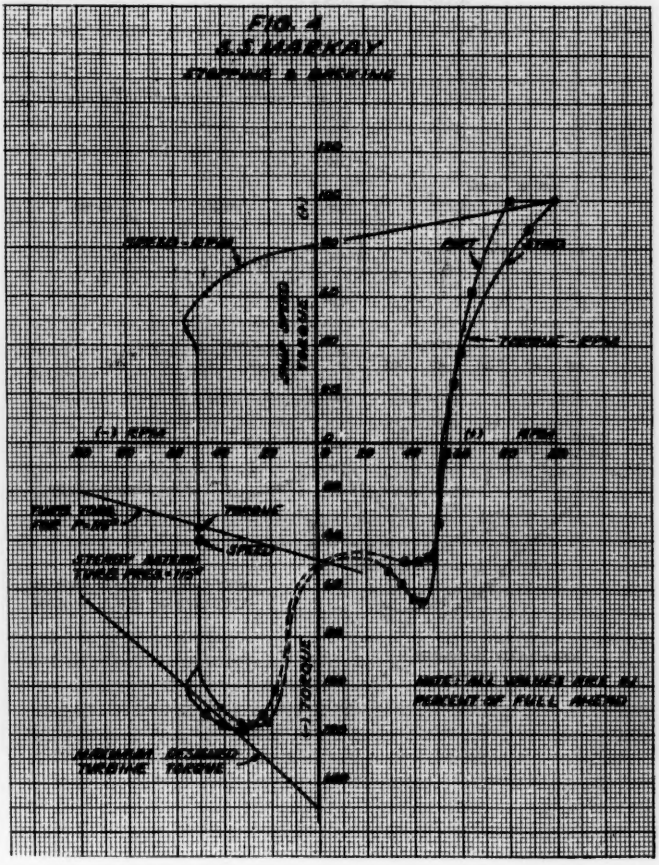
The deceleration curves of pitot tube and stadimeter or range finder are compared on Figures 1 to 3. The range finder data for the *Neosho* is not shown, since it is considered unreliable as speeds in excess of 20 knots were indicated. It will be noted in Figure 3 that the *Markay* did not attain full headway after the crash from full astern before the signal was given to close the throttles. However, for speeds between 4 and 10 knots, the deceleration is practically the same for both vessels. This is more fully compared in Figure 12.







The torque-RPM curves for the various ships are represented by Figures 4 to 8. The actual measured values of torque are plotted on RPM obtained from a curve of RPM on time. The speed-RPM relation shown on these figures is taken from curves of speed and RPM on time, and therefore the measured spots are not indicated.



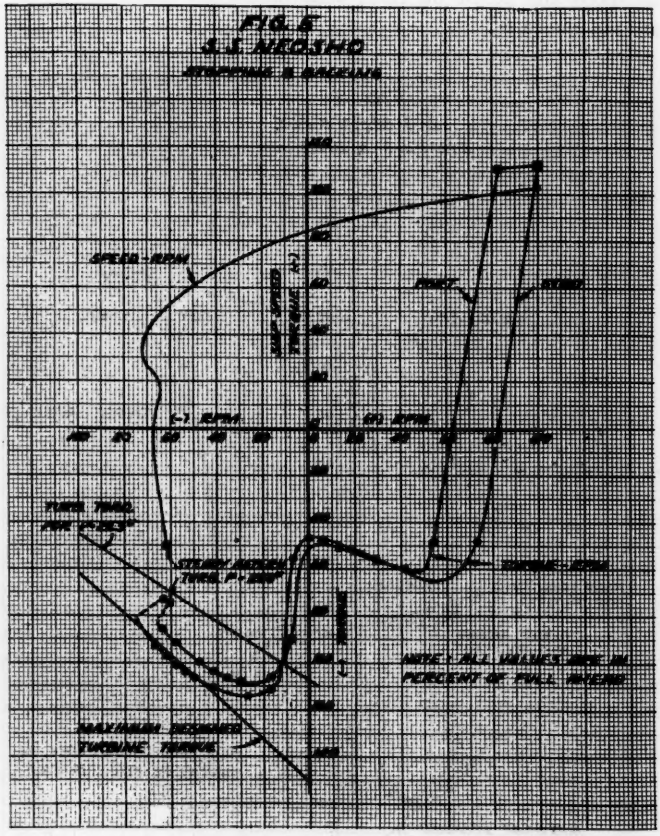


FIG. 5
A.S. NEOSHO
SPEED-RPM
PROP.
TURBINE
NOTE: ALL VALUES ARE IN
PERCENT OF FULL POWER
MAXIMUM DESIGNED
TURBINE POWER

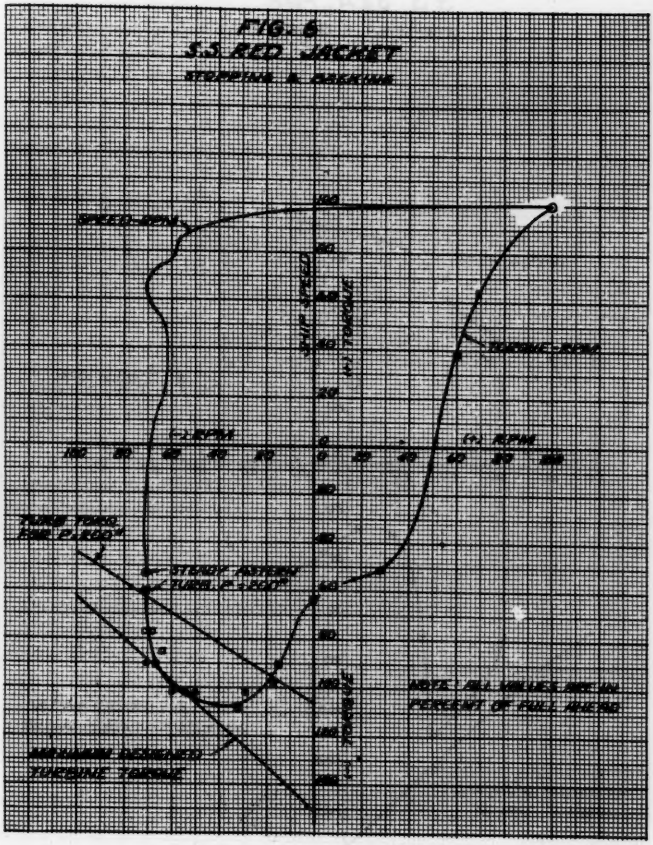


FIG. 6.
S.S. RED JACKET
STEAMING & MANEUVRE
SPEED-RPM.
THRUST-LBS.
TORQUE-FT. LBS.
THRUST-RPM.
TORQUE-RPM.
NOTE: ALL VALUES ARE IN PERCENT OF FULL HEAD

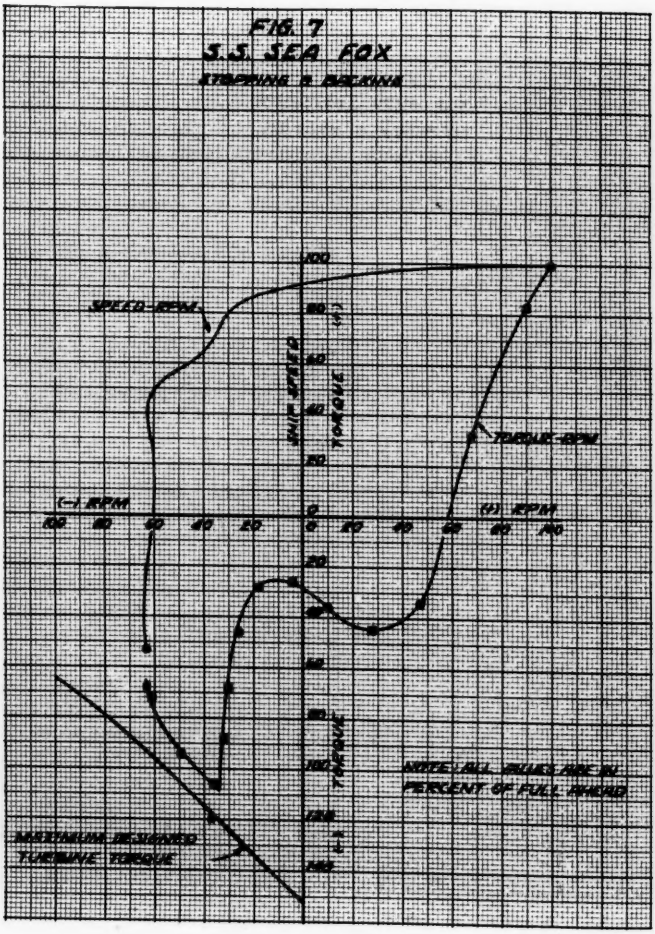


FIG. 7
S.S. SEA FOX
STEPHEN B. BALLANTINE
SPEED-RPM.
SHIP SPEED
TORQUE
TORQUE-RPM
(-) RPM
(+) RPM
NOTE: ALL VALUES ARE IN PERCENT OF FULL AHEAD
MAXIMUM DESIGNED TURBINE TORQUE

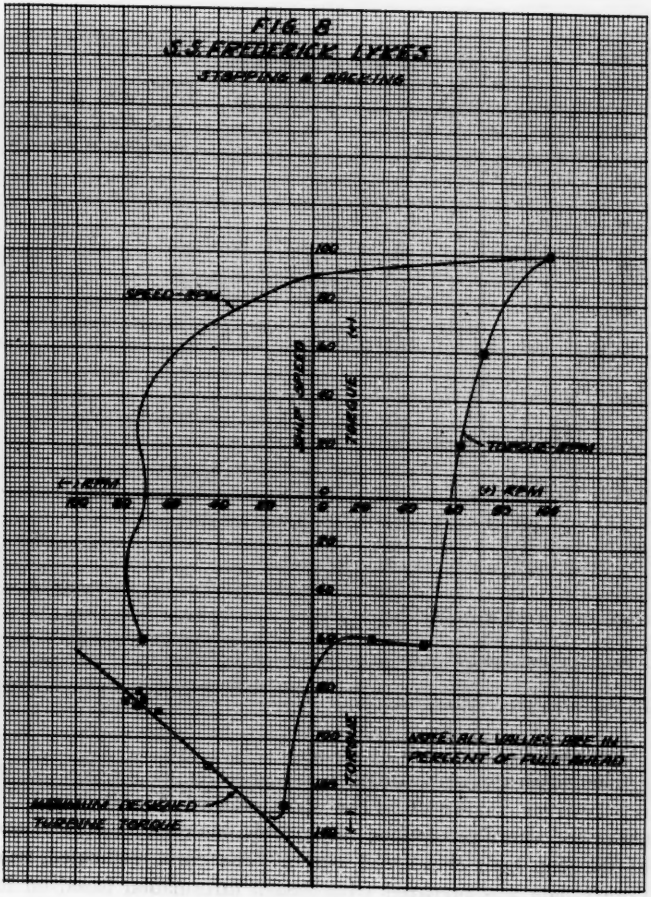
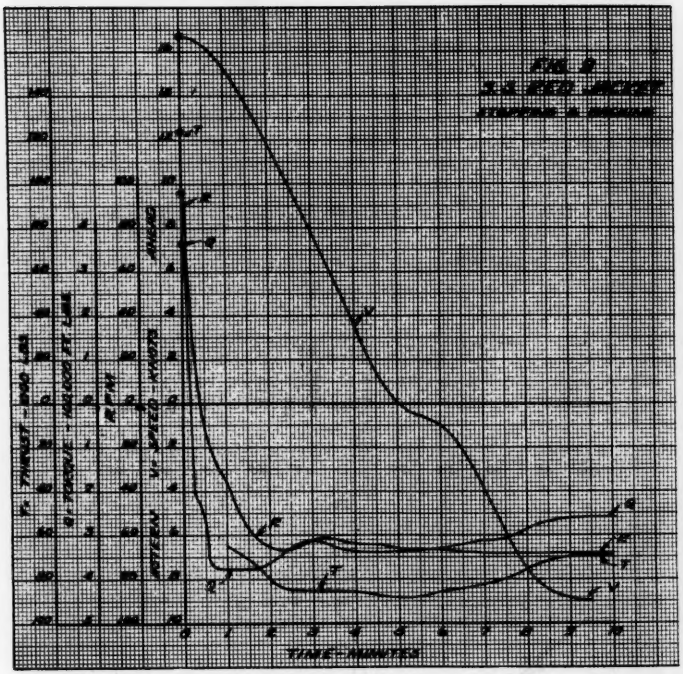


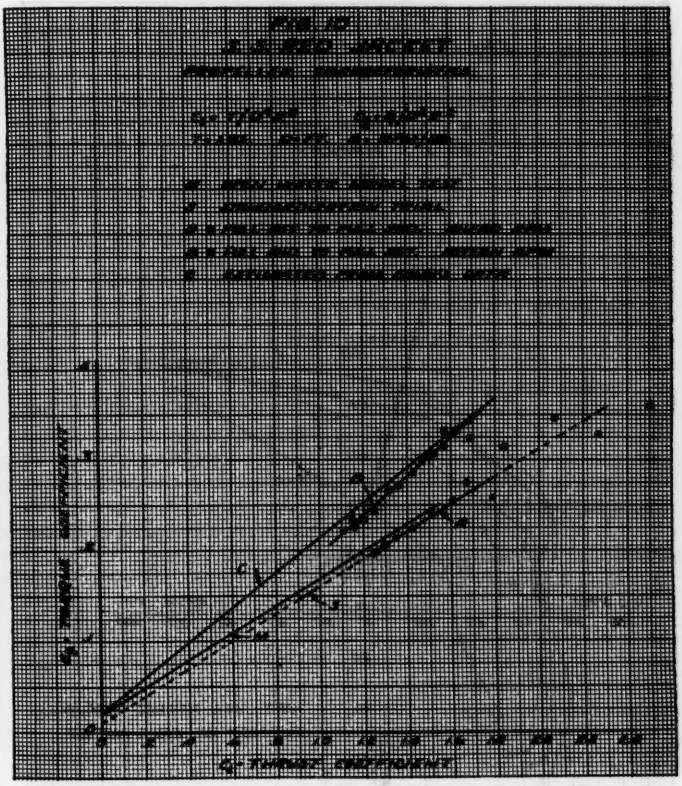
FIG. 8
S.S. FREDERICK LYKES
STRIPPING & BACKING
SPEED-RPM
TORQUE-RPM
TORQUE
TORQUE
NOTE: ALL VALUES ARE IN PERCENT OF FULL AHEAD
MAXIMUM DESIRED TURBINE TORQUE

Figure 9 represents the stopping data of the *Red Jacket* plotted on time. Thrust measurements during the first minute are not shown, since the meter could not be balanced rapidly.

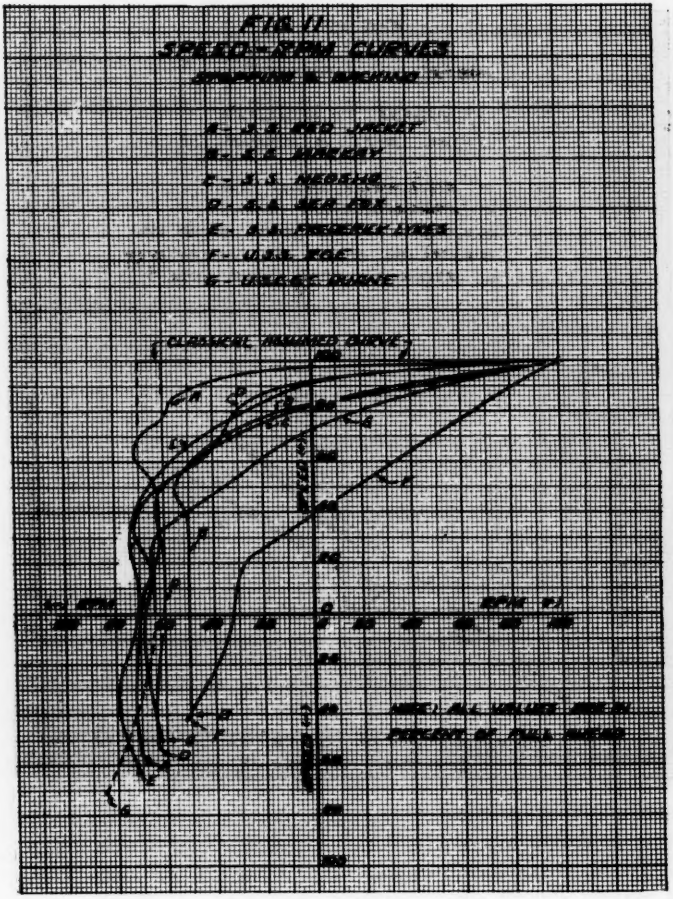


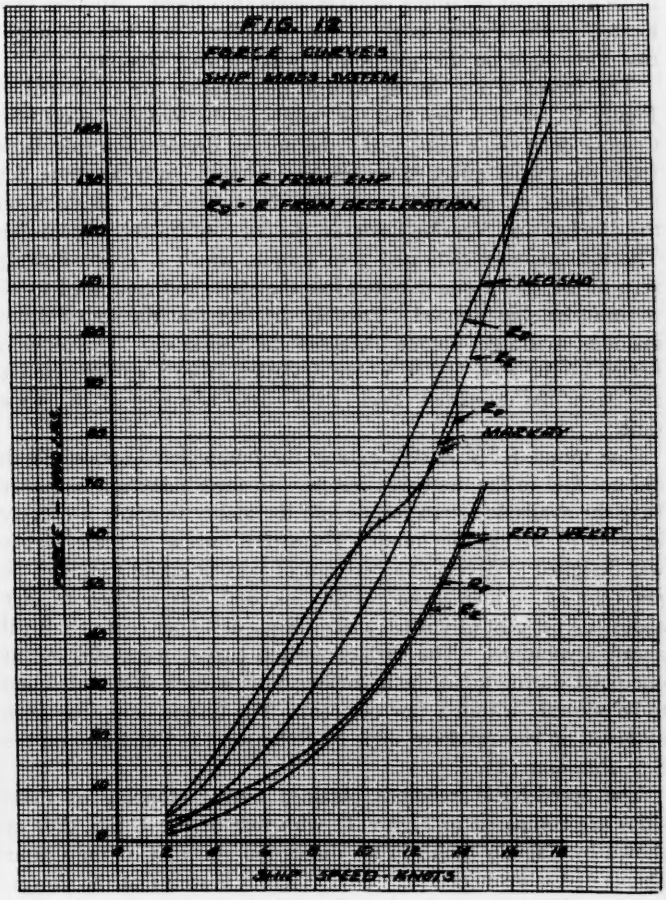
The thrust-torque relation of the propeller on the *Red Jacket* is shown on Figure 10. The spots marked A and B are for ahead and astern revolutions, respectively, but each includes motion of the ship both ahead and astern. The curve estimated from Conn's data was prepared from design information based on his tests. Briefly, his tests were analyzed in the Froude manner to determine effective pitch, thrust factors, initial and net torque factors, etc., for the various characteristics of form, section, number

of blades, area, thickness, rake, and pitch reduction which he tested. This design data was then used to predict the performance of the actual propeller when backing.



Figures 11 and 12 are the result of previous figures and require no explanation.





A short discussion of some of the results may be in order.

The deceleration curves obtained from the pitot tube readings generally were in agreement with those from the deck observations when corrections were made for current, the exception being noted above. It is believed that the pitot tube offers a simple and re-

liable method of obtaining instantaneous ship speed. Another tube located in the leading edge of the propeller shaft bracket might cast some light on the question of wake during stopping.

The forces acting during natural deceleration are compared in Figure 12, and it is seen that the results on the *Red Jacket* confirm Commander Gold's tests, that "the normal deceleration calculated from the model effective horsepower is sufficiently accurate for the purpose of estimating stopping characteristics." Results on the *Markay* and *Neosho,* however, do not confirm this conclusion. The torisionmeter indicated zero torque, which is approximately zero thrust, about one minute after the start, and continued to do so until the ship stopped. Possibly the fact that the ship did not maintain a straight course increased the resistance. This is confirmed by the *Red Jacket,* where deviation from a straight course and from the estimated deceleration curve occurred at the same time.

The torque-RPM curves exhibit the hump that Captain Smith did not find in the destroyer results. In these merchant ships we have the available astern torque and the ability to apply it quickly in order to produce the hump. The astern turbine of these vessels was designed to develop 85 per cent of normal ahead torque at 50 per cent of normal ahead revolutions, and was nozzled to give about 105 per cent of normal ahead torque at 50 per cent of normal ahead revolutions. The turbine torque curves were checked at one point during a half-hour backing test on each ship and found to be accurate within a few per cent. The torque hump has been observed on electric drive ships by Robinson (1)* and also on the S.S. *Potsdam* (2).

The speed-RPM relations obtained during stopping of these ships are compared in Figure 11 with that of the U.S.S. *Roe* as shown by Captain Smith in his discussion and that of the Cutter *Duane* of Commander Curry (3). Here it is seen that the slower, heavier ships approach the classical assumption of Admiral Robinson.

The thrust-torque relation obtained on the *Red Jacket,* is represented in Figure 10, which is another convenient method of comparing this relation. It is practically the same as TD/Q as

* Numbers in parentheses refer to bibliography at the end of this discussion.

used by Conn and referred to by Commander Gold. It is seen, however, that TD/Q ($= C_t/C_q$) is not constant over the entire range since C_t and C_q are not zero at the same time. Let us consider the reversal cycle on this plot. Starting at full speed ahead (about $C_t = 1.0$ on line S), C_t and C_q decrease, following a projection of line S and A. C_t and C_q become zero, then negative, and then reach negative infinity at zero RPM. At infinity, when the RPM reverses, the sign changes, and C_t and C_q become positive. As RPM increases, C_t and C_q decrease along line B, while the ship is brought to rest and accelerated astern, finally reaching steady astern (about $C_t = 1.1$ on line B). The plot shown here for the *Red Jacket* is very limited in scope. A larger range of test data (to a value of C_t backing of 4.5) can be covered by plotting the trial data of the *Clairton* (4). Treating Figure 38 of Commander Gold's paper similarly gives a range of C_t for ahead revolutions from plus 0.6 to minus 140, and for astern revolutions, from plus 1.0 to plus 85.0. These tests generally confirm the straight line relationship found for the *Red Jacket*.

The estimate based on Conn's data confirms the trial data approximately. This estimate is based on open water model tests, and naturally the flow conditions are not similar. It would be expected (as is the case) that the ship propeller would show a greater value of TD/Q under the actual flow conditions than in the open.

Conclusions to be drawn from these tests generally agree with those made by Commander Gold. It is hoped that Commander Gold will investigate the possibilities of the pitot tube and wake change if he undertakes further full scale trials.

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(2) Werft Reederei Hafen, January 1, 1936.

(3) H. H. Curry, "Stopping and Backing Tests—U.S.C.G. C. *Duane* and *Campbell*," JOURNAL OF THE A.S.N.E., Vol. 50, No. 4, Figure 8, on page 488; and Vol. 51, No. 1, page 109.

(4) R. R. Adams, "Thrust Measurements on the S.S. *Clairton*," Transactions of the Society of Naval Architects and Marine Engineers, Vol. 38, of 1930.

By R. G. Lorraine,† Charles Concordia † and
Ensign Donald L. Herr, * U.S.N.R.

At the time Commander Gold was carrying out his extensive stopping and backing trials on an actual destroyer, the U.S.S. *Roe,* the same problem was being considered in quite some detail from a theoretical viewpoint, using the most reliable ship data then available, for a contemporary design of a Diesel-electric drive.

This opportunity is taken to describe the method of analysis employed as well as to present results which qualitatively are entirely corroborative of those obtained by Commander Gold and discussed by Captain Lybrand Smith.

A. Ship and Propeller Equations.

The propeller and ship speeds and accelerations were calculated step-by-step from the equations of propeller torque and ship force. Commander Gold has already given these equations in his noteworthy paper but for immediate use they are reiterated here in somewhat different form.

The propeller torque equation is:

$$I_p \frac{d\omega}{dt} + T_p \pm T_f = T_e \qquad (1)$$

where

t = time in seconds;

ω = propeller speed in radians per second, ω being positive when the propeller is in forward rotation, negative for astern rotation;

I_p = electric motor and propeller moment of inertia in pound-feet-(second)^2;

T_p = propeller load torque in pound-feet, and is a function of ω, propeller speed and v, ship speed. It is obtained as T_p (ω, v)

† General Electric Company, Schenectady, New York.
* Bureau of Ships, Navy Department.

from experimental curves and varies for different hull-shapes and rated speeds.

T_f = propeller and motor-bearing function torque in pound-feet, assumed as a constant.

T_e = electrical torque in pound-feet, developed by the driving motor either as a function of ω, propeller speed, or as a function of motor slip, depending upon the mode of motor operation. T_e is obtainable from curves calculated for the various motor operating conditions.

The torque equation can be put in more general form by using per-unit quantities, which are denoted by primes, viz.,

$$a' = \frac{d\omega'}{dt} = \left(\frac{T_o}{I_p \omega_o}\right)\left(T'_e - T'_p \mp 0.02\right) = \text{Per Unit Propeller Acceleration} \qquad (2)$$

where

ω_o = base propeller speed in radians per second;

$$T_o = \text{base propeller torque in pound feet} = \frac{(\text{Motor HP})\ 550}{\omega_o}$$

The ∓ 0.02 is assumed 2 per cent frictional propeller torque and is of opposite sign to ω at any time.

For step-by-step calculation equation (2) and its integral are written in a more useful form as:

$$a' \Big]_{t+\frac{\Delta t}{2}} = \left(\frac{T_o}{I_p \omega_o}\right)\left(T'_e - T'_p \mp 0.02\right)\Big]_{t+\frac{\Delta t}{2}} \qquad (3)$$

$$\omega' \Big]_{t+\Delta t} = \omega' \Big]_t + a' \Big]_{t+\frac{\Delta t}{2}} \cdot \Delta t$$

This form assumes that the propeller acceleration at the half-time-interval is constant over the corresponding full-time-interval, an assumption which is valid for sufficiently small intervals.

The ship force equation is:

$$M_s \frac{dv}{dt} + F_v = F_p \qquad (4)$$

where

M_s = ship mass + mass of "entrained" water = (W_s/g) in pound-(sec.)^2 per foot;

v = ship speed in feet per second;

F_p = propeller thrust in pounds. This, like the propeller torque, is a function of v, ship speed and ω, propeller speed. F_p (ω, v) is obtained from experimental curves which differ for different hull-shapes and rated ship-speeds;

F_v = ship resistance in pounds, and is assumed proportional to the square of the ship speed, v.

Hence

$$F_v = F_o \frac{v^2}{v_o^2} \tag{4a}$$

where

v_o = maximum (base) ship speed in feet per second;

F_o = ship resistance, in pounds, at ship speed, v_o,

$$\text{giving: } F_o = \frac{(\text{Propeller Efficiency})\ (\text{Motor HP.})\ (550)}{v_o} \tag{4b}$$

Expressing per-unit quantities by primes, equation (4) becomes:

$$\beta' = \frac{dv'}{dt} = \left(\frac{F_o}{M_s v_o}\right)\left(F_p' - v'^2\right) = \text{Per-Unit Ship Acceleration} \tag{5}$$

As in the case of the torque equation, equation (5) and its integral are put in a form more convenient for step-by-step calculation, viz.,

$$\beta' \Big]_{t+\frac{\Delta t}{2}} = \left(\frac{F_o}{M_s v_o}\right)\left(F_p' - v'^2\right)\Big]_{t+\frac{\Delta t}{2}} \tag{6}$$

$$v' \Big]_{t+\Delta t} = v' \Big]_t + \beta' \Big]_{t+\frac{\Delta t}{2}} \cdot \Delta t$$

Equations (3) and (6) were used to determine theoretical ship and propeller behavior for various reversal procedures. Successive approximation of a' was avoided by writing the second equation (3) as

$$\Delta t = \frac{\omega' \Big]_{t+\Delta t} - \omega' \Big]_t}{a' \Big]_{t+\frac{\Delta t}{2}}}$$

and thus finding Δt from the known initial and assumed end values of ω', and the consequent value of a' thus obtained from the first equation (3).

PROPELLER TORQUE AND THRUST CHARACTERISTICS VS. SHIP AND PROPELLER SPEEDS. SLOW SPEED HULL OF U.S.S. JUPITER TYPE

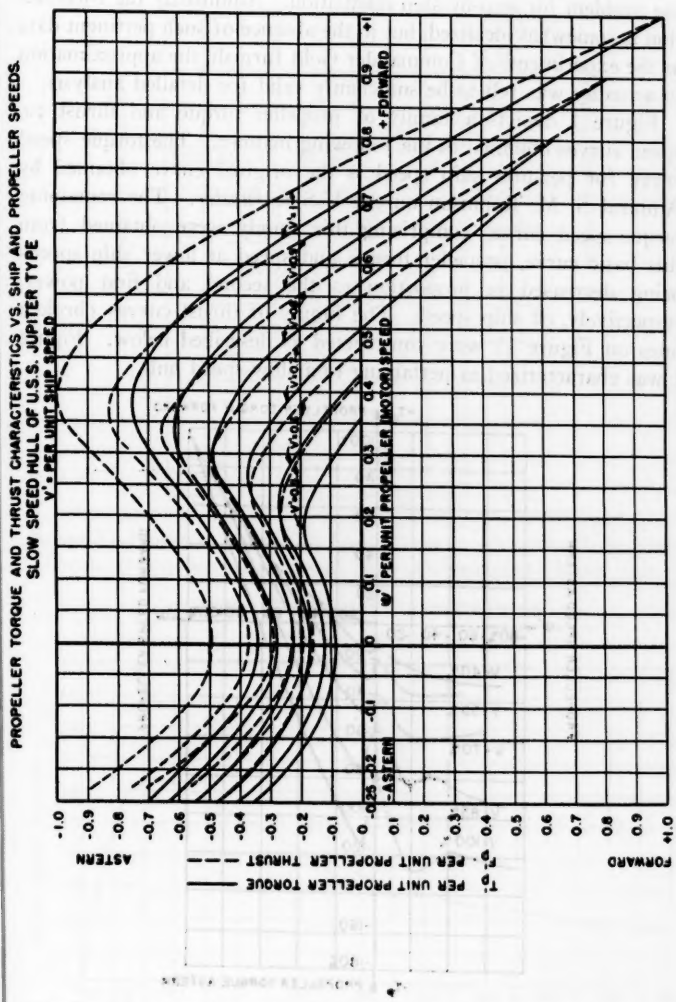
PROPELLER TORQUE AND THRUST CHARACTERISTICS VS. SHIP AND PROPELLER SPEEDS.
SLOW SPEED HULL OF U.S.S. JUPITER TYPE
V' = PER UNIT SHIP SPEED
ASTERN
FORWARD
PER UNIT PROPELLER THRUST
PER UNIT PROPELLER TORQUE
PER UNIT PROPELLER (MOTOR) SPEED
+FORWARD
-ASTERN

FIGURE 1.—PROPELLER TORQUE AND THRUST VS. SPEED CURVES FOR VARIOUS SHIP SPEEDS, CHARACTERISTIC FOR A SHIP OF THE U.S.S. "JUPITER" TYPE.

B. Torque-Speed Curve Families.

The foregoing section describes the theoretical formulation of the problem for step-by-step calculation. Admittedly the formulation is somewhat idealized, but in the absence of such pertinent data as the experiments of Commander Gold furnish, the approximation to actuality was felt to be sufficiently valid for detailed analysis.

Figure 1 depicts a family of propeller torque and thrust *vs.* speed curves obtained in the following manner. The torque speed curve for per-unit ship speed is the original curve obtained by Admiral S. M. Robinson on the U.S.S. *Jupiter*. The remaining torque-speed curves comprising this family were obtained from this basic curve, propeller torque and speed at lower ship speeds being decreased in proportion to the second and first power, respectively, of ship speed. The family of thrust curves (broken lines on Figure 1) were constructed as described below. Figure 1 was characterized as pertaining to a slow-speed hull.

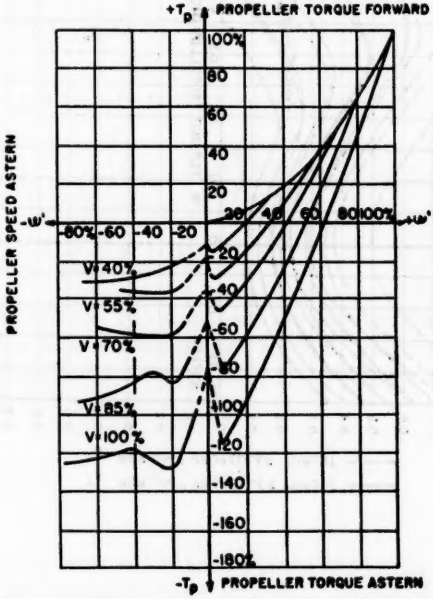

Figure 2.—Propeller Torque vs. Speed Curves for Various Ship Speeds, Characteristic for Fast-Speed Hull—Determined by Model Tests.

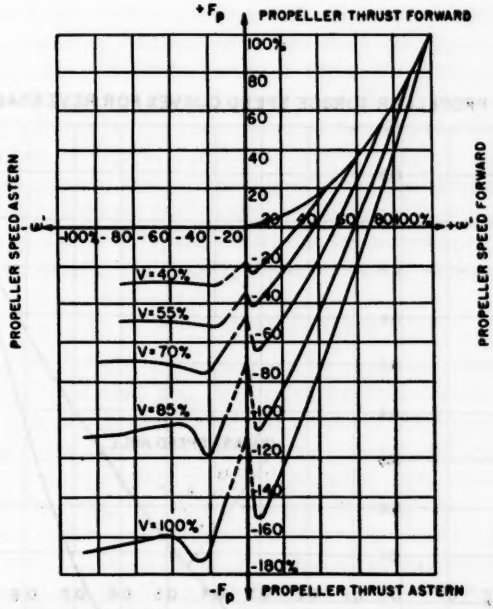

FIGURE 3.—PROPELLER THRUST VS. SPEED CURVES FOR VARIOUS SHIP SPEEDS, CHARACTERISTIC FOR A FAST-SPEED HULL—DETERMINED BY MODEL TESTS.

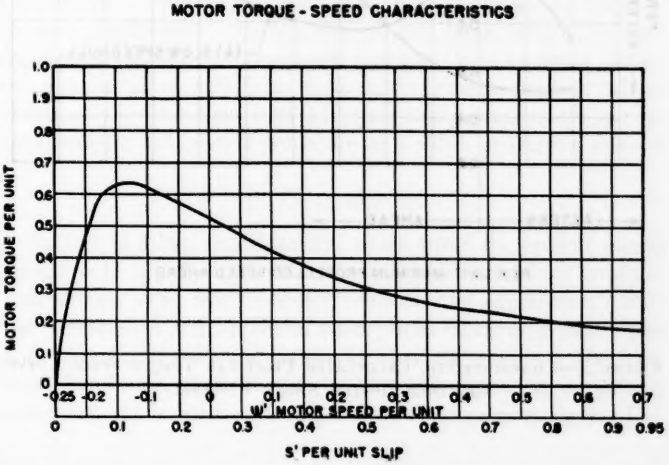

FIGURE 4.—TYPICAL CALCULATED ELECTRIC MOTOR TORQUE-SPEED CURVE.

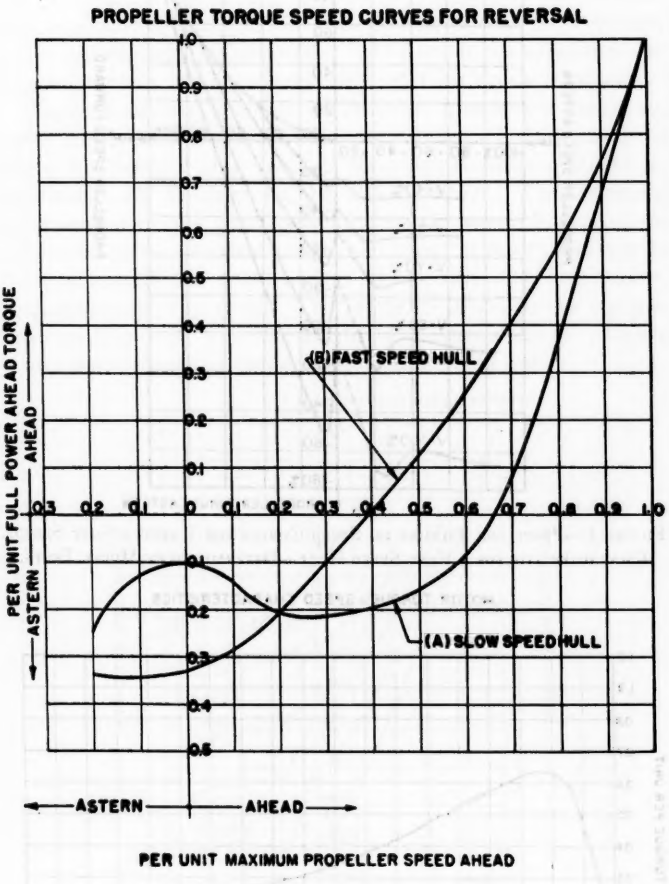

FIGURE 5.—CHARACTERISTIC CALCULATED PROPELLER TORQUE-SPEED CURVES OBTAINED DURING SHIP'S REVERSAL.

Figures 2 and 3 are the torque and thrust *vs.* propeller-speed curves obtained by Mitzlaff on model tests corresponding to a fast-speed hull. The thrust curves of Figure 1 for the slow-speed hull were derived from the torque curves of Figure 1 by preserving as nearly as may be the same relations between the torque and thrust curves of Figures 2 and 3.

Ship resistance is taken proportional to the square of the speed and is based on 65 per cent (assumed) efficiency at full speed.

Figure 4 shows a typical electric motor torque-speed characteristic and supplies the quantity T_e mentioned in section A above.

C. Calculated Results.

Making use of the formulas given by equations (3) and (6) above, the theoretical ship and propeller behavior for various reversal procedures were calculated by step-by-step methods. A small portion of the results obtained are depicted in Figure 5.

Curve (A) represents a typical propeller torque-speed curve calculated by using the slow-speed hull characteristics of Figure 1 and the motor characteristic of Figure 4. Curve (B) is a similar curve calculated by using the fast-speed hull characteristics of Figures 2 and 3 and the motor characteristic of Figure 4.

It is to be noted in both cases that these calculated results fully corroborate in a qualitative manner the experimental results of Commander Gold's experiments and those discussed by Captain Smith. Particularly, the absence of any large torque "hump" during reversal for the horsepower employed is to be noted. Such information is of very great value to the design engineer whose job it is to determine with efficiency and economy the power requirements and control procedure of a ship's propulsion system.

D. Conclusions.

Subject to minor modifications, which make the present results appear only slightly optimistic, the analytical procedure produces qualitatively valid results, amply confirmed by experiment. Further refinements and additional assumptions concerning the torques and forces acting do not materially alter the results. The technique employed, that of step-by-step calculation, is admittedly laborious, but is justified in a problem of large magnitude. Dif-

ferential-analyzer solution of the pertinent differential equations could be resorted to in future investigations in order to obviate materially the expenditure of time and labor involved in such computations. This is particularly true when variation of one or more of the parameters, ship mass, nature of ship resistance, hull or motor characteristics, is considered.

As an alternative method of attacking the same fundamental problem, that of understanding the reversal characteristics of a large ship, it is hoped that the above procedure suggests new possibilities of investigation.

The writers are grateful to Captain Smith and Commander Gold for this opportunity of discussion.

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(2) George Mitzlaff, "Electric Propulsion—Development in Electric Propulsion of Ships-Manoeuvering", Shipbuilder and Marine-Engine Builder, April, 1939.

FINAL DISCUSSION.

By Lt. Commander P. D. Gold, Jr., U.S.N.,
Author of the Original Paper.

Captain Lybrand Smith's Discussion.

The writer concurs with Captain Smith, that model tests are not entirely satisfactory for obtaining stopping and backing data, however, model tests are very helpful in establishing trends and in comparative studies. Due to the low Reynold's number at which model tests necessarily are conducted, it is, even in the case of the normal hull resistance and self-propelled tests, necessary to make corrections for this Reynold's number, or friction effect. Once a number of different full scale trials have been compared with model tests, it is believed that then such tests will be satisfactory for obtaining stopping and backing data. It is believed that corrections for cavitation may be applied to backing tests with equal accuracy as now applied to self-propelled model tests. In either case the conditions in the variable pressure tunnel do not represent a condition similar to that of the ship, for as stated by Captain Smith, the tunnel is not large enough to hold the model hull, and the model propeller is not subject to the turbulence and variations in wake existing on the full scale ship.

There has been considerable discussion in the past over the so-called virtual mass. It is undoubtedly true that the boundary layer has a forward velocity which affects both the ship and the propeller. The effect on the ship may be represented as an added mass as suggested by Dr. Kempf. It might equally well be considered to be a decrease in resistance. In any event, it is well within the errors of other necessary assumptions and may be neglected until the major inaccuracies can be eliminated. As wake fraction and thrust deduction are for steady state conditions known only approximately, a small change created by the boundary inertia effect would be difficult to measure and it is thought that this

effect also may be neglected, as it is well within the order of accuracy of the final results. From studies of boundary velocities of ships, it is difficult to understand how the boundary layer effect could be as great as 16 to 20 per cent even if it be assumed that all of the boundary layer inertia must be counteracted. The 5 per cent found by Dr. Kempf is believed to be closer to the actual effect.

That stopping and backing trials be conducted on one of the electric drive battleships is an excellent suggestion. Accurate measurement of input torque by electrical means, the large torque available, the tip submergence of the propellers and the large inertias involved, create an ideal combination for such a test.

Lt. Commander Labberton's Discussion.

Lieutenant Commander Labberton has made a very interesting analysis of the *Roe* trials, expressing the results non-dimensionally. The torque coefficient has been expressed as a function of slip alone. In the case of the destroyer, air was drawn from the surface. In other cases propeller cavitation may exist. Under these conditions the relations between torque coefficient and slip become indeterminate. Also the neglect of the effect of wake introduces errors of indeterminate magnitude. The writer cannot therefore, agree with the statement that "These tests (of the *Roe*) indicate certain fundamental phenomena which, when put in basic form, may enable us to compute with great accuracy the performance of any propeller in stopping a ship when a known torque is exerted in reverse."

The usefulness of dimensional analysis for the recording of test data is dependent upon similitude. Where similitude does not exist the writer can see no advantage in expressing propeller characteristics in this manner.

While the assumption of a constant wake fraction may lead to errors of small magnitude, it should be recognized that, due to the importance of the relative velocity of the water to the blade that any calculation of stopping characteristics of ships appreciably different from those already tried will be, at best, approximate.

The writer still believes, as stated in the original paper, that the results of the trials of the *Roe* can be applied accurately only to ships of similar hull and propeller characteristics.

Mr. D. C. MacMillan's Discussion.

Mr. MacMillan has in his discussion added valuable data to the small amount now available. Considering the vast difference between the destroyer and these ships of the cargo class, the agreement in results is exceptionally good. Mr. MacMillan states that the substitution of cameras for human observers and recorders renders the data more accurate and reliable and that the use of shore stations directed by radio to determine the instantaneous ship position during deceleration is more reliable than range finders on ship board. Thanks to Commander Curry's excellent discussion of the limitations of the shipboard instruments for determining ships position these instruments, although first considered, were soon discarded. An excellent substitute was already available. The Experimental Model Basin had developed instruments and technique for the determination of instantaneous ships position for turning trials, therefore this procedure was not originated for the trials under discussion.

It is of interest to note the appreciable difference in time in stopping the shaft, the ship, and the head reach between identical ships. If this is indicative of operational variations, it would appear that this operational variable is the most important variable of all. Analysis of a number of trial reports of naval vessels leads to a similar conclusion. It is the writer's belief however, that the differences are usually more instrumental than operational.

That the use of model resistance for determining normal deceleration is at best approximate, is concurred in. However, from the results of the trials of the *Neosho* and *Markay* it is apparently sufficiently accurate for the purpose of estimating stopping characteristics since it is within the order of accuracy of the trials themselves. The calculated $\frac{dv}{dt}$ from the model test would indicate that the ships resistance during the normal deceleration trials

was less than the trials indicated, but the comparison between the two trials considered indicates nearly equal error between the trials themselves

Speed	Model	*Neosho*	*Markay*
8	.7	1.0	1.1
10	.98	1.4	1.35
12	1.47	1.8	1.5
14	2.12	2.2	2.0
16	2.62	2.7	—
18	3.32	3.6	—
19	3.93	—	—

The installation of a pitot tube for measuring ships instantaneous velocities was considered for the Destroyer trials, however, due to the short time available to prepare for these trials, it was decided to depend upon the instantaneous position curve obtained by the shore parties. The expressed hope that in the future the velocities be obtained directly, by means of a pitot tube, is echoed by this writer. The influence of currents is of such importance during the stopping period of heavy ships that pitot tube velocity measurements are practically necessary if great accuracy is to be attained.

With reference to the curves shown on Figure 10, the Ct, Cq relation agree very well with the *Roe* trials. It will be noted that for curve B, the crash astern condition, $\frac{Ct}{Cq}\left(\frac{TD}{Q} \times \frac{P}{D}\right)$ is practically constant at five. " T " being the propeller thrust uncorrected for thrust deduction. The statement that "it would be expected that the ship propeller would show a greater value of $\frac{TD}{Q}$ than would the model propeller in open water, is not understood. In fact, abnormal disturbances usually tend to decrease efficiency, which being equal to $\frac{T}{Q} \cdot \frac{V}{2\pi N}$ would, for constant $\frac{V}{2\pi N}$ indicate a lower $\frac{TD}{Q}$ for the condition of the propeller behind the ship.

The curves shown on Figure 11 form an interesting comparison. The slope of these curves depends not only upon power, but upon propeller characteristics and ship's mass. The point at which the curves cease to be fair depends primarily, in the opinion of the writer, on submergence and blade width. Considering these added variables it is believed to be erroneous to consider these curves to represent the effect alone of "power available."

Mr. R. G. Lorraine, Mr. Charles Concordia and Ensign D. L. Herr's Discussion.

The method of calculating the variables involved during the stopping and backing process, as described by Mr. R. G. Lorraine, Mr. Concordia, and Ensign Herr, is the best the writer has seen to date. While somewhat tedious the results are well worth the effort. The possible inaccuracy of such a calculation should be continuously borne in mind, however, and sufficient motor torque should be available at all conditions of motor slip to allow for these inaccuracies.

In the original paper, the writer chose the symbols now in common use by both marine and air propeller designers in this country. These symbols are the same as defined in Admiral Taylor's classic "Speed and Power of Ships" and adopted by the National Advisory Committee for Aeronautics. While the writer also, prefers those symbols chosen by Ensign Herr and Messrs. Lorraine and Concordia, that is "F" for a force whether it be resistance or thrust and "T" for torque, it is believed that less confusion results from the use of those symbols in common usage.

Inasmuch as the value of such calculations as presented is dependent upon the accuracy of the assumptions, these assumptions will be considered.

(a) "R or ship's resistance is proportional to V^2." This resistance varies over wide ranges depending upon speed and hull characteristics. For the *Roe*, the resistance varied from approximately V^2 at low speeds to $V^{3.6}$ at high speeds and therefore an assumption of V^2 would have led to errors of large magnitude. It is believed preferable to calculate the resistance, either from

model tests or from Taylor's Standard Series. An approximate logarithmic curve may then be drawn through this curve so calculated.

(b) Propeller efficiencies assumed are subject to large errors, however no alternate method is suggested. The possible errors should be recognized and ample "factors of safety" used. The use of $\frac{TD}{Q}$ as suggested by J. F. C. Conn, appears to the writer to be preferred to the use of the word "efficiency" which has little meaning at zero shaft speed or zero ship speed.

(c) The neglect of the effect of wake and thrust deduction is acceptable since the order of accuracy of the calculations does not demand such refinement. The fact that a large thrust deduction is undoubtedly present should, however, be recognized.

(d) The assumption of a constant friction of 2 per cent is certainly acceptable. In fact friction effects may be neglected as being small compared with other inaccuracies.

The writer wishes to express appreciation to the authors of the discussions and also to the Federal Shipbuilding and Drydock Company, and to the General Electric Company for permitting submission of such valuable data.

RESEARCH.

ADDRESS BY

THE HONORABLE HERBERT HOOVER AT THE COMMENCEMENT EXERCISES OF HAVERFORD COLLEGE, HAVERFORD, PENNSYLVANIA, JUNE 7, 1941.

It is a great pleasure at last to be able to accept an invitation to come to Haverford. And for two reasons. It was founded and has been sustained by the faith in which I was reared. And when I was yearning to go to college my stern Quaker managers insisted that my spiritual welfare would be unsafe in a nonsectarian institution. They therefore secured for me a scholarship in this college and an alternate one at Earlham. But my ambitions then were in sciences and engineering—and Haverford, at that time, was not strong in that branch. That deficiency has long since been overcome and Haverford now turns out its quota of first quality men in those branches.

THE NEED FOR IMMEDIATE EXPANSION OF RESEARCH.

I wish to take this occasion to urge again the immediate expansion of our research facilities in pure and applied science, and the more systematic application of what we already know in these fields. That is at all times the highway to increased efficiency and increased productivity. And this action is the more urgent today because through it we can contribute to alleviate the hardships of our American people both during and after this war.

However, as Huxley insists, let me first define my terms. For the practical purposes of this discussion, pure science research means the search for fundamental natural law. That is the search for truth. That is a good exercise any time.

Applied science research is the application of such discoveries to practical use. That is invention. Pure science is thus the raw material of applied science.

For this discussion national industrial efficiency means producing more goods and services per capita.

I do not want you to think that I am building a rigid wall between pure and applied science. Their purposes and borders are much blended. I might mention another distinction between them. There are no big money-rewards to the pure scientists. Sometimes they get an obituary notice on the fourth page inside.

I am not one who believes that the whole of civilization is founded upon scientific discovery and the inventions. I could complain of a thousand economic and social ills that have come from the industrial revolution, including the increased capacity for mass-murder. But because of this war I can also tell you that unless we quickly have more of this same discovery and invention and a more efficient application of what we already know our standard of living and even our civilization will degenerate.

First—Whatever the outcome of these present wars, there is one thing certain—we shall have to divert a large segment of national energy to armament for years to come. That at once decreases the output of articles for public consumption. Unless we can by some parallel action increase our production of consumption goods, it means an immediate reduction in our standard of living. And there is nothing that can so contribute to make up that deficiency as new discovery in pure science and new inventions. From that source we can get more labor-saving devices, and better methods which increase production. We can get new materials to substitute for those we have to take from the people for munitions. Nor do we need depend wholly upon new discovery and new invention to increase our national productivity. We can organize to better apply what we already know—and that can produce immediate results.

Second—There is another and immediate purpose of enlarged research. The airplane has revolutionized warfare. It has made the aggressor far more powerful. It has made small nations helpless to defend their liberties. It has made the butchering of women and children a part of war. The world has not yet found the defensive answer. The airplane was born from the science of

physics. The answer might come from there. And it imperatively demands research and more research—and at once. For if that answer could come and come quickly it would turn the whole fate of the world.

Third—Whatever the outcome of this war one more thing is certain—this whole world and our own country will be greatly impoverished and smothered with debt. There is only one certain road to rapid recuperation. That is to increase the technological power of the nation.

It is increased productivity at lower costs that the nation will need. In the past we have always reaped such advances from important discovery, invention and organized application.

There is, of course, the commonly proposed idea that we could increase production and alleviate scarcity in goods by longer working hours. But it is far better that we accomplish this by more scientific discoveries, new labor-saving devices, new methods, or the more vigorous organization and use of what we already know.

I do not need to be told all the old arguments that invention puts men out of jobs. That ghost is as old as when Eli Whitney invented the cotton gin. That ghost should have been laid when it saw the effect of the gas engine. For every man in the livery stable yesterday there are 20 in garages today.

Nor is there justification for that other truculent ghost which wails about more pressure upon individual workers. It only means that we work our machines and heads harder.

How Much Do We Expend Upon It?

Our pure science research is dependent largely upon our universities and a few specially endowed institutions. Their research work is decreasing rather than increasing. The reduced interest return on the endowments of our institutions is driving them to cut into research rather than teaching. I doubt that even without curtailment their total resources for pure science research amount to $20,000,000 a year. Thus we allow pure science a national expense account of about 5 per cent of what we allow for cosmetics.

We make a better showing in applied science. Governmental and industrial research, including agriculture, is supplied with probably $200,000,000 a year. That is not equal to our cosmetics allowance by a good deal.

The Process of Discovery in Pure and Applied Science.

And now under the pressure of preparedness as we need more new methods, more new inventions, more new labor-saving devices, and more new materials, let me explore the processes by which we get them.

I have said elsewhere that there was a time when invention came from the starving genius in the garret. We got the steam engine that way. These gifts now come from long years of patient experiment in great laboratories. Like the growth of plants cell by cell, of fact upon fact, some day there comes forth the blossom of discovery, the illuminating hypothesis or the great generalization. And finally it fruits into a multitude of inventions. Moreover, poverty does not essentially clarify thought. Nor does it provide laboratory equipment. Bread and water diet has been discarded by the mother of invention.

I can easily illustrate the movement of pure science from discovery to invention. Our electrical industries today are the result of search into pure science. They are the result of a half-century of step-by-step advance in the realms of pure physics and mathematics. It is from these realms that Faraday extracted the transformation of mechanical energy into electricity through induction. It is an old story that Gladstone was induced to visit Faraday's physics laboratory to see this new scientific contraption. When Gladstone, a practical man, inquired whether it would be of any use to mankind, Faraday opined, "I think some day you will be able to tax it." But it was long years afterward that Edison Thompson, Siemens and the other engineers translated Faraday's discovery into power and light. And today it moans and groans with taxation. But it pays the taxes.

And taxes are not the major achievement of electricity. Rivers of sweat have been saved from the backs of men. Watt and Faraday did more to reduce the 84-hour week to a 44-hour week and to give more good jobs to men than all the laws and all the organizations which have been embattled in this cause.

Infinite drudgery has been lifted from the hands of women. The electric light alone has relieved the human race from the curse of always cleaning oil lamps, scrubbing up candle drips, and everlastingly carrying one or the other of them about.

And its benefits are not all economic. They lift the spirit. It adds cheer to life—enables us to postpone our spectacles for a few years longer. It has made reading in bed infinitely more comfortable. By merely pushing a button we have introduced the element of surprise in dealing with burglars. The goblins who lived in dark corners and under the bed have now been driven to the outdoors. The doctors now peer into the recesses of our insides. Our electric light enables our cities and towns to clothe themselves in gaiety by night, no matter how sad their appearance may be by day. It has lengthened the hours of our active lives and enabled us to read the type in the telephone book. It has become the friend of man and child.

But now to return to my illustration of the progress of skilled research in electrical physics. From Faraday we could move on to Maxwell's formulation of the electrical wave theory by pure mathematics. Hertz confirmed Maxwell in experiment and proved the radiation of waves through the air. It was the engineers Marconi and DeForest who transformed these discoveries of pure physics into the radio communication. Out of that we get jazz by night and war scares by day. Between them it diverts our minds from other woes.

The Possibilities of the Future.

And now somebody will say that all this is in the past. What of the future? Are there any great fields to explore from which such benefits can spring again? The answer is that never in the whole history of fundamental science have there been so many vistas opened before us as right now. Discovery and invention expand in geometrical progression. Inventive minds play upon every new invention and breed a dozen more.

Much of our progress comes from the invention of new instruments—that is, new tools for research. They are born from long hypothesis and prior experiment. Out of the discoveries in atomic structure and radiation have come new tools, new instruments, which open still wider avenues for research.

By use of these laboratory tools or instruments comes a multitude of discoveries and inventions. Every time we get a new one we have some addition to human progress. And in the very recent

past our scientists have evolved a number of these instruments, the possibilities of which stretch our imagination to the breaking point.

The harnessing of the cathode ray through the cathode tube was the key to television. And a great industry has been born. From that application the cathode tube was further developed to the astonishing new electron microscope. Today we have photographs of red blood corpuscles as large as dish pans. We are actually able to photograph some molecules. A dime could be enlarged to a mile in diameter if we possessed a big enough film.

And from all of this we will know more of how to limit the conduct of bacteria and increase the energies of men. We will know far more about cellular structure and the make up of molecules which will improve our industrial processes. The field of discovery with this instrument has hardly been entered. There are only two or three of these miscroscopes in use. Shortly there will be scores of these in the hands of our scientists—provided someone furnishes the money to build and operate them.

And there are the cyclotron and the mass spectrograph, and their opening of the whole vistas of atomic structure. Actual atomic power here has been experimentally produced.

All these instruments are revolutionary. All advance in laboratory instruments is not so sensational. Minor ones happen every day which contribute to progress. The spectroscope was for years just a tool for the astronomer. Today it has been transformed into an instrument for accurate metallurgical analysis to control our steel furnaces and thus the methods of industry improve.

The invention of the steam engine did not appear in the headlines of a world then engaged in the Napoleonic Wars. Yet it saved the world from much of the impoverishment of those wars. History may yet record that the headlines of today should have been devoted to the inventions around the electron rather than war news.

Again It Needs Money.

And now I come again to the question of men and money for these explorations. I believe every physical scientist in this country today could name new paths, new fields, that await for the money to mobilize men and equipment. Many of their results may be

sterile but somewhere something comes—an improved instrument, a new path opens and in the end an increase in the standard of living.

Someone will say that these great discoveries of fundamental law cannot be forced—that we must wait for them. I do not contend you can go out and buy such a genius and have him produce every morning. But I do say that he cannot perform without equipment and support.

Pushing the Application of the Science We Already Know.

Nor do we have to wait for pure science research to give birth to new laws, new instruments, new methods, new power, and new labor-saving devices. In many directions we sorely need support for research in the application of pure science discoveries we already know.

One of the greatest of our problems right now is to develop more industrial raw materials which our farmers can produce in substitution for their overproduction of food. I doubt whether we are spending five millions a year looking for them. And we are compelled to subsidize the farmer with a billion a year and to regiment him besides until we find some such solution as this.

And in the realm of industry there are further vast possibilities in synthetic fibers, rubber, in the plastics, or new sources and methods of making and use of cellulose and a score of other things. There is the field of metallurgical treatment of low-grade ores. If we developed such methods we could free ourselves from depending upon imports of chrome and manganese. We need substitutes for materials needed for defense. I doubt that actually three millions a year are being spent in such research laboratories.

Elimination of Waste.

Nor to improve our national efficiency do we have to wait for even these better applications of science.

Under the pressures of emergency we can increase national efficiency through the elimination of existing industrial waste of motion and materials. That requires that we remove every sort of restriction both by capital and labor which impedes or penalizes the use of better methods and better machines or the working of these machines full time.

At once someone will rise up and say that these are fine generalities; that they lack an exact and particular action which will bring results right now while we are in the middle of this preparation for defense. If you will go back to the last war and examine the methods which we then adopted for the elimination of waste, the simplification of methods, the substitution of materials for continuous operation, you will find an immediate and enormous field already pioneered. If the armed forces need to take 20 per cent of our shoes by such devices we can at once increase the output of our existing factories and thus save a cramp in our supply of shoes.

Some Other Economic Consequences.

We will spend 20 billion a year on armament. We will pile most of it up in national debt. If we would invest 1% of this in an insurance policy called research and elimination of waste, we might save some of the bankruptcy when these wars die.

It is possible that the aggressive superiority of the airplane in war may be checked. And that becomes vital to the freedom of nations and to stop the massacre of women and children.

And it is not beyond human imagination that the quiet, unobtrusive scientist and engineer in their laboratories might make all nations self-contained in raw materials and their production of goods. And what a holocaust that would be for international quarrels and international thinking—and for much of the causes of war.

Some Social Consequences.

And that brings me to another phase. We have been told with monotonous repetition by the collectivists and left-wingers that our frontiers are gone. They say our industrial plant is built. They claim there is no safety valve for human energies. They assure us that we have come to an age of humdrum problems of under-consumption, over-production, and the division of the existing pot. They say that new opportunity for youth has shrunken.

That is not so. There was never in history a more glorious frontier for youth than today. Adventure and opportunity beckon in every avenue of science. They beckon from the great profession of men trained to research. They beckon from its thousands of

applications. From it spring tens of thousands of new services and industries. In them human courage, character, and ability have an outlet that never came even with the two-gun frontiers. Just as the new villages followed the stockades of the frontier, so do new cities follow every new mastery of technology and power.

And let those who lament the loss of frontier life not forget the adventures along every mile of highway and even with the traffic cop. That all emanated from the discovery of the combustion engine.

Some Spiritual Consequences.

But research can bring far more than defense from aggression, or the opening of new frontiers, discoveries, adventures, inventions, labor-saving devices, more power or increased standards of living. There also lies in these fields a contribution to the moral and spiritual welfare of mankind. Here is the lifting of men's minds beyond the depressing incidents of the day. Here lies the unfolding of beauty, the ever-widening of the boundaries of knowledge. Here is the "inculcation of veracity of thought" in a world sodden with intellectual dishonesty. Here is the harmonizing of the individual to the pattern of his environment. Here is the confirmation of a Supreme Guidance in the universe far above man himself.

And today we need more of these things to help save and build a great nation.

NOTES

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SYNCHRONOUS MOTORS.

This article, by F. K. Brainard, of the A. C. Engineering Department, Allis Chalmers Manufacturing Company, is reprinted from Product Engineering for May, 1941. It deals principally with considerations of load and inertia effects.

In the larger sizes of alternating current motors the synchronous type has a number of advantageous features. Its starting characteristics can be made suitable for nearly any constant-speed drive; when running it can improve a low plant power factor; it has high efficiency; and for low-speed drives its price is comparatively low. But when selecting a synchronous motor care must be exercised to select one with starting as well as running characteristics suited to the particular machine to be driven. Conversely, the necessary starting characteristics of a motor are determined by the power required by the driven machine during the starting period and by the inertia (flywheel effect or WR^2) of the rotating parts. As shown later, the selection of suitable characteristics of a proposed synchronous motor may depend to a considerable extent upon the inertia of the load. From the standpoint of starting it is always desirable to reduce the inertia as much as possible, but sometimes it is necessary to increase it to obtain satisfactory operation under load.

A synchronous motor is quite similar in construction to a synchronous A.C. generator. Alternating current power is delivered to the armature and mechanical power is taken from the shaft, instead of vice versa, as is the case with a generator. Direct current excitation is supplied to the field winding in either case. Either the field or the armature may be the revolving element, but like the generator and for the same reasons it is usually built with the field revolving.

Before a synchronous motor can operate normally it must be brought into synchronism with the supply circuit, connected to it, and fully excited. The usual method of doing this is to equip the pole faces with an amortisseur winding, which is similar to the squirrel cage winding of a squirrel cage induction motor, and to start by connecting the armature to the A.C. power supply, the field circuit being closed through a suitable resistance. The alternating current in the armature produces a magnetic field which revolves around the periphery of the armature at the rated speed of the motor. This revolving magnetic field induces currents in the amortisseur winding and in the closed field circuit, the reaction of which upon the magnetic field produces the torque of the motor during the starting period. Then after the motor has reached a certain speed, generally about 95 per cent of full speed, direct current excitation is applied to the field winding, and the motor will go to full speed and will carry loads up to the pull-out torque of the motor.

Obviously, the torque developed by the motor during the starting period must be enough to bring it sufficiently close to synchronism to enable it to pull into step when direct current excitation is applied; and this will depend upon the power required by the driven machine at this point, and upon the other factors mentioned later in this article.

Figure 1 shows the speed-torque curve of a certain synchronous motor and the centrifugal pump which it drives. The pump cannot be unloaded for starting. As is seen, the torque developed by the motor greatly exceeds that required by the pump until 97 per cent speed is approached, at which point the torques are equal. Direct current excitation is applied at about this speed and the motor pulls into step readily.

The most desirable shape of speed-torque curve depends upon the requirements of the driven machine. A high-resistance amortisseur winding results

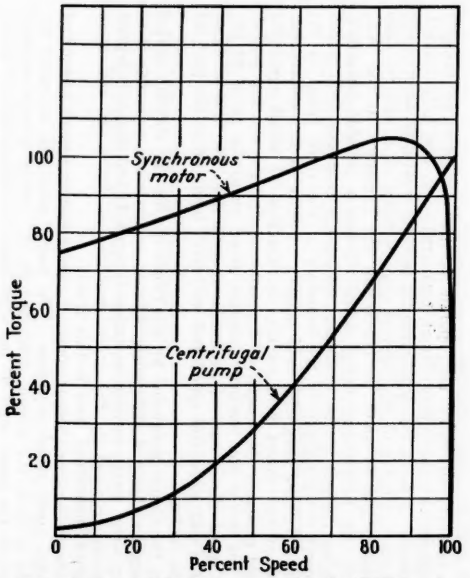
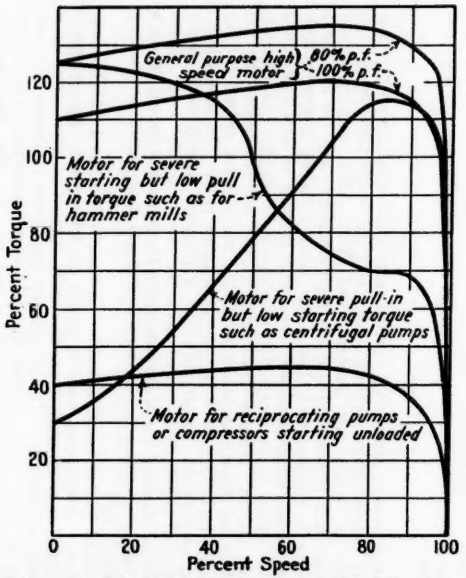


Fig. 1—Starting as an induction motor under the influence of an amortisseur winding and the closed field circuit, the speed of a synchronous motor builds up to about 95 percent of synchronous speed. At this point d.c. excitation is applied to the field and the rotor pulls into step with the revolving magnetic field produced by the field current and reaches full synchronous speed, at which it continues to run at any load up to the maximum pull-out torque of the motor which is usually 150 percent or more. Fig. 2—Typical speed-torque curves of motors suitable for various kinds of service



in relatively high starting and low-pull-in torques, while a low-resistance amortisseur winding results in low starting and high pull-in torgues. Figure 2 shows typical speed-torque curves of motors suitable for various kinds of service.

Because of its effect upon the power system and upon the size and cost of the motor, it is desirable to reduce the load as much as possible during the starting period. Frequently it can be materially reduced quite easily as is the case with most reciprocating compressors, positive pressure blowers, and centrifugal pumps or blowers. Some machines, however, such as cement mills, cannot be unloaded for starting. Others, such as mine fans, which have large rotating parts can be partially unloaded. Still others, such as induced draft fans having heavy rotating parts cannot be unloaded very much. Each of these cases requires careful consideration, particularly those where both large starting loads and heavy rotating parts are involved. If the starting and pull-in torque requirements are severe it may be desirable to use a mechanical, magnetic, or hydraulic clutch of some kind between a standard synchronous motor and the load. This is sometimes less costly and more satisfactory than a special motor suitable for use without the clutch. In this case the motor will be started light, brought up to speed,

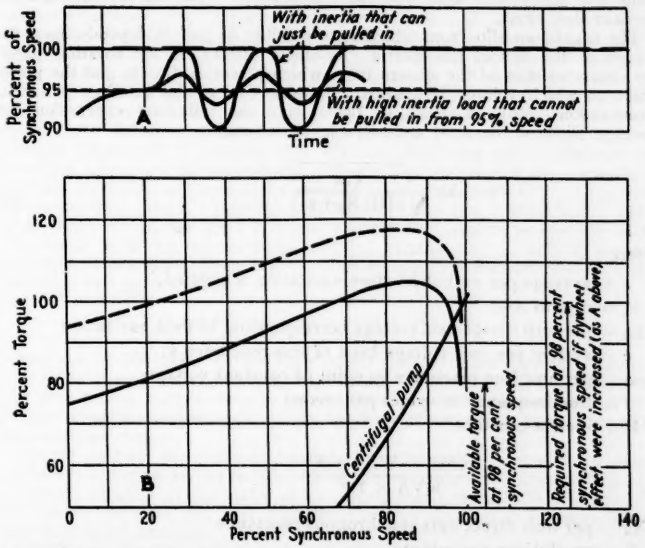


Fig. 3—Diagram A illustrates approximately the effect of the inertia of the load on the ability of the motor to pull into step. When d.c. excitation is applied to the rotor, pulsations are set up as shown. If the inertia is within proper limits the motor will pull into step and run at 100 percent synchronous speed when the proper phase angle is reached. If the inertia is too great the pulsations will never reach 100 percent speed. To do so it would be necessary in this example to run the motor on the starting winding up to 98 percent synchronous speed before applying d.c. excitation. If applied to the pump in Fig. 1 it would have the effect shown at B. The motor design would have to be changed to get approximate characteristics shown on the dotted line

and fully excited before engaging the clutch. Care must be exercised to see that the motor has enough pull-out torque to prevent its being pulled out of synchronism when the clutch is engaged.

Excessive flywheel effect manifests itself in two distinct ways, both of which make starting more difficult.

The first is to increase the total energy loss in the amortisseur winding during the starting period and consequently the heating. This is because the acceleration is lower and the high amortisseur loss persists for a longer period. In an extreme case both armature winding and armortisseur winding might be damaged.

The second effect is the reduction of the pull-in torque of the motor. After reaching the highest speed at which it will operate as an induction motor, the application of D.C. excitation introduces an additional component of torque which alternates at slip frequency. This torque causes pulsations in speed above and below the mean, magnitude of which depends upon flywheel effect, other things being equal. If the flywheel effect is increased the pulsation in speed will be less, and the initial speed at the time that excitation is applied must be increased if the motor is to pull into step, because these pulsations in speed are the means by which the gap between induction and synchronous operation is bridged. The curves in Figure 3 explain this effect.

The maximum slip from which it is possible to pull in depends upon the design of the motor, the inertia (fly-wheel effect) of the rotating parts, the characteristics of the system from which power is drawn, and the phase angle at which voltage is applied. Shoults, Crary, and Lauder (A.I.E.E. Transactions, Vol. 54, 1935, p. 1385) give the following expression for average slip for the most unfavorable phase angle:

$$s < k_2 \sqrt{\frac{E_d e}{\pi f H (X_d + x_1)}} \qquad (1)$$

where

s = average per unit slip before excitation is applied,

k_2 = 0.55 to 0.60

E_d = per unit direct axis voltage corresponding to field excitation

e = system per unit voltage back of line reactance x_1

x_1 = per unit line reactance to point of constant voltage

f = line frequency in cycles per second

H = inertia constant

$$= \frac{0.231 \times WR^2 \times N^2}{KVA \times 10^6}$$

X_d = per unit direct axis synchronous reactance

N = revolutions per minute

As shown by Equation (1), the maximum permissible slip at the time excitation is applied varies inversely as the square root of the flywheel effect. This means that if the flywheel effect is large, it becomes necessary to bring a given motor closer to synchronism than would otherwise be necessary; and the load must be correspondingly reduced.

In order to obtain a basis for the comparison of the relative flywheel effects of various loads at various speeds the N.E.M.A. has adopted values of "normal load WR^2" based upon the empirical expression:

$$\text{normal load } WR^2 = 3.75 \times \frac{10^5 (\text{hp})^{1.15}}{N^2} \qquad (2)$$

This expression represents a value of inertia of driven load that can generally be pulled into step without difficulty if the torque load is not greater than the "nominal pull-in torque" as defined below.

Pull-in torque is the maximum constant torque under which the motor will pull its connected inertia load into synchronism. Equation (1) shows that this depends upon the flywheel effect of the load and the characteristics of the power system, as well as upon the design of the motor itself, and cannot be determined unless these quantities are known. However, it has been found in nearly all cases that 5 per cent slip is permissible; that is, if the motor is designed so that it will reach 95 per cent speed as an induction motor when developing the specified torque, it will usually pull in when excitation is applied. For this reason "nominal pull-in torque" has been defined by N.E.M.A. as the torque at 95 per cent speed when operating as an induction motor without excitation.

By assuming an adequate power system and by making certain other approximations, the ratio of "nominal pull-in torque" to "pull-in torque" can be expressed as a function of the horsepower and power factor ratings of a motor and the ratio of the WR^2 of the load to the "normal load WR^2." This is shown by the curves in Figure 4, and is useful for approximate work. If accurate results are desired, however, Equation (1) should be used.

As an example of the use of the above curves, let us assume that it is desired to drive a certain induced draft fan with a 300 HP., 80 per cent power factor, 1200 RPM. synchronous motor. During normal operation with the exhaust gases hot it requires 275 HP. to drive the fan. Dampers are provided to reduce the load during starting, but because of the necessity of starting cold the load cannot be reduced to less than 225 HP. at the close of the starting period. The flywheel effect of the fan is 9000 lb.-ft.^{2} which is 49 times the normal load WR^2 for the 300 HP. motor, as given in tables of normal WR^2 for synchronous motors. Values for A and B are 2.37 and 0.90, respectively, from Figure 3. The required nominal pull-in torque will then be approximately:

$$225 \text{ HP.} \times 2.37 \times 0.90 = 480 \text{ H.P.}$$

Therefore, a 300 HP., 80 per cent power factor motor having a nominal pull-in torque of about 160 per cent instead of the N.E.M.A. standard 125 per cent pull-in torque must be used. This illustrates how important it is to give careful consideration to the flywheel effect of the load in applying synchronous motors.

So far we have been considering only the detrimental effects of the inertia of the revolving parts, but in many cases it is necessary to increase the inertia to obtain satisfactory operation of the synchronous motor-driven unit. This is true in spite of the fact that the synchronous motor is essentially a constant-speed machine.

Additional flywheel effect is frequently used to advantage to reduce the variations of power and current drawn from the line which would otherwise result. If the load is a periodically pulsating one, such as that of a reciprocating pump or compressor, these variations would frequently be serious if suitable values of flywheel effect were not used. Excessive variations of current result in corresponding variations in voltage which may cause objectionable flickering of lamps supplied from the same circuit. In extreme cases damage to the motor is possible. The American Standards Association rule covering this is as follows:

"When the driven load such as that of reciprocating type pumps, compressors, and others require a variable torque during each revolution, the combined installation shall have sufficient inertia in its rotating parts to limit the variations in motor armature current to a value not exceeding 66 per cent of full load current.

"The basis of determining the variation shall be by oscillograph and not by ammeter readings. A line shall be drawn on the oscillogram through the consecutive peaks of the current wave. The variation is the difference between the maximum and minimum ordinates of this envelope. This variation shall not exceed 66 per cent of the maximum value of the full load current of the motor. (The maximum value of the motor armature current to be assumed as 1.41 times the rated full load current.)"

In many cases the percentage variation in power and current drawn from the line is less than the percentage variation in torque causing it. This is not always the case, because there may be partial resonance between the frequency of certain impulses and the natural frequency of the unit on the power system.

Driving a compressor and its flywheel from a large power system by means of a synchronous motor is analogous to driving it from the same prime mover through gearing and a somewhat flexible line shaft. It runs at an average speed proportional to the speed of the prime mover, but there may be momentary variations in speed above and below the average because of the flexible connection. If the prime mover is large compared with the compressor, it is easy to see that the line shaft and flywheel in the mechanical analogy constitute a torsional pendulum which will oscillate at a definite frequency determined by the moment of inertia of revolving parts and the stiffness of the line shaft. If the compressor produces impulses the frequency of which is close to this natural frequency, partial resonance occurs and violent oscillations may result. In this case the variations in torque of the line shaft will be much greater than the variations in torque of the compressor. If these oscillations are very great, the line shaft in the mechanical analogy might break, which corresponds to the synchronous motor dropping out of step. The torque per electric radian, P_r is analogous to the stiffness of the shaft.

The A.S.A. formula for natural frequency of a synchronous machine on a large system is:

$$F = \frac{35,200}{N}\sqrt{\frac{P_r \times f}{WR^2}} \qquad (3)$$

where

P_r = synchronizing power in kilowatts per electrical radian as defined in A. S. A. rules

f = frequency of the system in cycles per second

N = rated speed in RPM.

WR^2 = flywheel effect of unit in pound-feet^2

A factor has been adopted by the N.E.M.A. for use in determining the flywheel effect needed for various compressors. Designated by C in Equation (4), it provides a means for comparing the relative values of flywheel effect of units of various sizes and speeds

$$C = \frac{0.746 \times WR^2 \times N^4}{P_r \times f \times 10^8} \qquad (4)$$

Values of C corresponding to 66 per cent power pulsation have been adopted by N.E.M.A. and A.S.A. for various types of compressors having various

methods of unloading (A.S.A. Standards C-50-1936). By solving Equation (4) for WR^2 the following convenient expression is obtained:

$$WR^2 = \frac{1.34\ P_r f C}{(N/100)^4} \qquad (5)$$

STANDARD TORQUE CHARACTERISTICS OF SYNCHRONOUS MOTORS.

MOTOR SIZE	POLES	TORQUE (PERCENT OF RATED FULL-LOAD TORQUE)		
		Starting	Pull-in	Pull-out
General Purpose				
Up to 200 hp (1.0 power factor).....	4	110	110	150
	6–14	110	110	175
Up to 150 hp. (0.8 power factor).....	4	125	125	200
	6–14	125	125	250
Large High Speed				
250–500 hp. (1.0 power factor).....	4–14	110	110	150
250–500 hp. (0.8 power factor).....	4–14	125	125	200
600 hp. and larger (1.0 power factor)....	4–14	85	85	150
600 hp. and larger (0.8 power factor)....	4–14	100	100	200

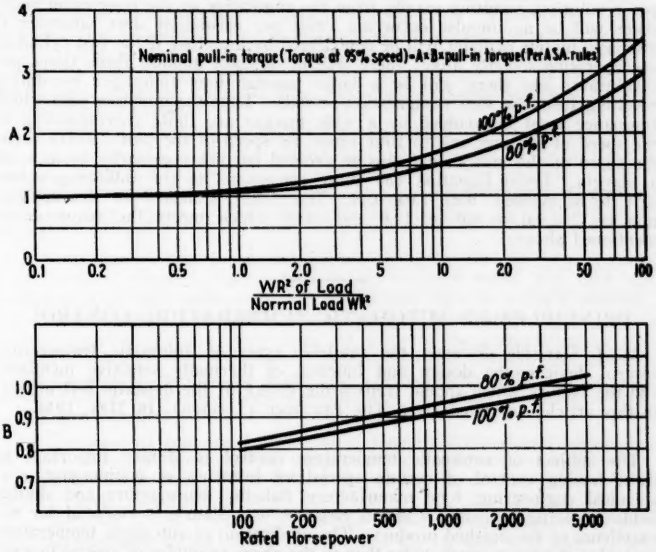


Fig. 4—These curves are useful for determination of the ratio of "nominal pull-in torque" to "pull-in torque" of a motor when WR^2 of driven machine is known

By using values of C from the A.S.A. rules in Equation (5), the limiting values of WR^2 are readily obtained. From this it is seen that the flywheel effect necessary for any compressor driven by a synchronous motor drawing power from a large system varies directly as the compressor factor C, the frequency of the system, and P_r; and inversely as the fourth power of the speed. Since P_r varies almost directly with the rated output of the motor, the WR^2 will also, other things being equal.

The permissible values of C for a given compressor depend almost entirely upon the shape of the tangential effort curve. This can be shown as follows:

Solve Equation (3) for WR^2 and substitute this in Equation (4). We then obtain:

$$\frac{N}{F} = \sqrt{\frac{C}{9.25}} \qquad (6)$$

The ratio of speed to natural frequency thus depends upon C, and since the ratio of electrical power variation to torque variation for each harmonic of the torque curve depends principally upon the ratio of speed to natural frequency it will depend upon C also. Frequently a judicious modification of the design of the compressor will make a material difference in the permissible values of C, and for this reason close cooperation between the compressor and synchronous motor designers is often advantageous and desirable.

As an illustration of the frequencies to be avoided, consider a horizontal duplex double-acting ammonia compressor with cranks at 90 degrees. This compressor will have an appreciable unbalanced impulse occurring once per revolution, resulting mainly from the angularity of the connecting rods; there will be no impulse occurring twice per revolution since whatever is produced by one cylinder will be neutralized by the other if the two cylinders are operating exactly alike; there will be a small impulse three times per revolution; and there will be a large impulse four times per revolution resulting from the four compression periods. Thus the impulses with which resonance must be avoided by a wide margin are those corresponding to the speed of the unit and to four times the speed of the unit. Three times the speed of the unit should also be avoided but not necessarily by as wide a margin. From Equation (5) these correspond to the following values of C to be avoided: 9.25, 1.03, 0.58. The A.S.A. standard for this application is C = 2.0 to 6.0 or 12.0 and over, which meets the requirements mentioned above.

PRINCIPLES OF AUTOMATIC TEMPERATURE CONTROL.

M. J. Gartside discusses the available types of automatic temperature control devices, the design and function of thermally sensitive members, and the factors which should control the choice of thermostatic instruments in this article, reprinted from The Engineer (London), 16 May, 1941.

The subject of automatic temperature control is already important to those having control of certain specialized branches of engineering, *e.g.*, chemical engineering, food manufacture, bakelite manufacture and similar fields involving processes where a constant temperature is essential for the excellence of the finished product. The application of automatic temperature control is, however, much wider than to the above examples of special process work and no engineer can have failed to deplore at some time, either consciously or unconsciously, the lack of temperature control, even if it only be in relation to his own personal comfort.

In considering the subject it is important to obtain first of all a completely clear conception of the justification for automatic temperature control. Without this clear conception one may be led to the erroneous application of temperature control in an attempt to solve a problem which is in fact soluble only by correct application of basic heating principles.

Consider the heating of any body, whether it be liquid in a process tank, the mould of a bakelite press, or the air in a factory workshop. The body will receive heat from whatever source that may be chosen and will lose heat due to radiation, convection, and conduction; in many applications heat will also be removed by removal of a portion of the heated body and replacement by fresh cold material. At the commencement of the heating process all the heat input will go to raise the temperature of the body; as the temperature rises the heat losses due to the first three factors just mentioned will increase and the rate of rise in temperature will decrease until finally a point is reached where the losses balance the heat input and a steady temperature results. Theoretically, this latter position only obtains after infinite time, but for all practical purposes it can be considered to take place after a finite time which will depend upon the rate of heat input, the rate of heat loss, and the capacity for heat of the body considered.

Assuming that all the conditions remain constant, a steady temperature will be obtained; if such an assumption were correct there would be no need for temperature control. However, in practical heating applications many factors obtain which prevent the above-mentioned ideal conditions from being fulfilled, and so make necessary means of control. It is clear that increased input or decreased losses will cause an increase in temperature, and *vice versa,* and in practice it is very seldom possible to ensure that either the heat input or losses are constant. Furthermore, in many applications heat exceeding the amount necessary to overcome the losses at the required temperature has to be provided at frequent intervals to give quick increase of temperature to the required value; it is often inconvenient to have the increasingly retarded rate of temperature rise which is the characteristic obtained when the heat input just balances the losses at the required final temperature.

Basic Principles.—Generally speaking, a thermostat, or temperature control device, consists essentially of an element, sensitive to temperature, which is located in the medium to be controlled. The sensitive element transforms changes in temperature into changes of mechanical dimension which, in turn, cause one of the following functions to be performed indirectly:—

(i) Reduction of heat input immediately to zero when the temperature of the sensitive element slightly exceeds the required value, and then re-establishment of heat input when the temperature falls by a slight amount. This is conveniently termed an "on-off" system of control.

(ii) Gradual decrease of heat input as long as the temperature of the sensitive element exceeds the required value by a small amount, and increase of heat input when the opposite is the case. Such a system, the increase and decrease in heat input normally being effected at a constant rate, is described as "floating" control.

(iii) Decrease of heat input with rise of temperature and *vice versa,* the heat input being proportioned to temperature difference above and below the required temperature. This system is termed one of "modulating" or "proportioning" control.

The Meaning of the Term "Differential."—From a consideration of the above an important factor comes to light and that is that it is only by an actual change in temperature of the medium taking place and being transmitted to the thermostat that the latter can operate to effect the change of heat input necessary to counteract the change in temperature.

This leads to an important definition, namely, the "differential" of the temperature control system. With an "on-off" system the differential is the temperature difference between the temperature at which the sensitive element causes the heat input to be reduced to zero with rising temperature, and that at which it causes it to be resumed with falling temperature. With a modulating system the differential is the difference between the temperature at which no heat input is given and that at which full input is given, between which temperatures heat input is varied proportionally.

It will be noted that in considering the above care has always been taken to refer to changes in the temperature of the sensitive element, and not to changes of the temperature of the medium being controlled. The differential referred to above has always been the theoretical or operating differential.

Actually, the temperature of the medium being controlled may or may not be the same as that of the sensitive element, and, what is an exceedingly important practical point, the reading of a thermometer immersed in the medium may be yet again different. Factors causing this discrepancy are:

(i) Difference in position of the sensitive element and the thermometer coupled with a temperature gradient between these two positions in the body being controlled.

(ii) If the medium being heated is air, it may be that the thermometer and sensitive element are either subject to different amounts of radiant heat or the nature of their surfaces may differ, resulting in their receiving and reflecting different amounts of heat.

(iii) When change in temperature occurs, the temperature of the sensitive element (and of the thermometer) will always change at a slightly later time than the temperature of the medium, since time is taken for heat to be transmitted from the medium to the sensitive element, and *vice versa*.

Under what may be termed ideal conditions for temperature control, rates of change of temperature are slow, and temperature is even throughout the bulk of the medium heated. Transmission of temperature changes to the sensitive element of the thermostat is sufficiently rapid in comparison with the time required for further change of temperature to take place in the medium that the temperature of the sensitive element may be said to follow exactly the temperature of the medium controlled.

The differential measured by a thermometer in the medium will then be identical with the operating differential of the thermostat.

In practice conditions differ widely from the above and to some degree, at any rate, it is usual for the change in temperature of the thermostat to lag behind the change in temperature of the medium controlled. When this occurs it is clear that the thermostat will operate only after a change in the temperature of the medium larger than the differential of the thermostat has obtained. A similar effect may be experienced due to the heating means provided having a large heat capacity. For example, for heating an oven, heavy electric elements may be employed, which, even though switched off by a thermostat, still retain sufficient stored heat to impart a further rise in temperature to the air in the oven.

In such circumstances, which obtain to a greater or lesser degree in almost every temperature control application, it will be seen that the actual differential obtained and measured by a thermometer will always exceed the theoretical differential of the temperature control device.

The Use of Modulating Control.—In certain circumstances rates of temperature change may be so rapid, when full heat input is provided or heat input reduced to zero, that no practical temperature control could follow the change.

A case in point is air-conditioning work in which a continual stream of air passes through a heating battery of low heat capacity. Shutting off the heat supply completely will cause an immediate rapid fall in the temperature of the issuing air.

In such a case "on-off" control of the heat input is unsatisfactory; full heat input must be provided only when the temperature of the medium is well below the required temperature; heat input must be gradually reduced with increase of temperature near the control point, and the heat input must be reduced to zero only when the temperature is well in excess of the required temperature. This requires the use of a modulating system of control.

Heat Acceleration.—Consideration of "on-off" control, and the difference between the theoretical differential of a thermostat and the actual differential obtained in practice, suggests that if by any means the sensitive element of the thermostat could be made to receive heat from the source more directly than by conduction from the medium being controlled, then the increase in differential due to the inherent lag of the sensitive element could be avoided. This is often arranged in practice either by placing the thermostat relatively close to the heat source, so that it receives heat directly, or, as is done with electrical thermostats, particularly for the control of air temperature, by providing a separate electric heater in the thermostat, the heater being energized when the thermostat is calling for heat.

Such special arrangements have to be made with great care, as unless the additional heating effect given to the thermostat is kept within certain close limits the effect may be to give effective control of the temperature of the heating source, and hence constant heat input, rather than control of the medium.

Design of the Thermally Sensitive Member.

The sensitive element of any temperature control is invariably a matter for exceedingly careful consideration by the manufacturer, and the attributes which are important are, first, that its physical properties are such that when located in intimate contact with the medium to be controlled it shall respond as quickly as possible to changes of temperature in that medium; and secondly, that such changes of temperature shall provide mechanical movement and force to give the maximum possible amount of work for indirectly performing one of the functions mentioned earlier.

In ordinary commercial temperature control work, the thermally sensitive element of temperature controls can be divided into the following two classes:—

Bi-metallic Systems.—Bi-metallic systems are those in which force and movement are obtained by virtue of the difference in expansion of two dissimilar metals. Usually, the arrangement consists of a tube of brass or other metal having large expansion coefficient in respect of temperature and a rod of a nickel-iron alloy having an exceedingly small expansion cofficient; alternatively, the same two metals may be used rolled together in the form of a strip, the expansion of one side of the strip with increasing temperature causing the strip to bend and enabling work to be performed.

In a bi-metallic system of the rod and tube type, the movement obtained with change of temperature is small—a brass tube 12 inches long expands only 0.001 inch for a change in temperature of 10 degrees F. To offset this the force available is exceedingly large.

Filled Systems.—A filled system consists of a sealed bulb, which forms the sensitive element proper, and which contains a liquid having either large volumetric expansion or large change of vapor pressure with change of

temperature. The sealed bulb is usually connected by capillary tubing to a sealed metallic bellows and changes in temperature at the bulb result in a corresponding change in pressure and volume of the charged system, so making the bellows expand or contract. Movement of the bellows against a spring causes operation, according to temperature, either of an electric switch or of a valve directly.

In some types of instrument, *e.g.*, for room temperature control, the sealed bellows expansion chamber is actually the sensitive bulb itself, no capillary tube being involved.

Function of the Thermally Sensitive Member.

In the above consideration of the design of the thermally sensitive member, it will have been noted that variation of temperature produces force and movement, *i.e.*, an amount of work which can be used to control the heating means in a number of ways, chief of which are the following:—

(1) One of the most common ways is the operation of an electric switch. The electric switch may control electric heating loads directly or through the medium of a relay; it may switch the electric current to a magnetic or motor-operated hot water or steam valve, so providing "on-off" control to whatever means of heat input is employed; or it may similarly control a motor operating to open or close dampers on a boiler, for example.

The design of the electric switch is exceedingly important, since if control of temperature within close limits is required, the mechanical work available to operate the switch is limited to a small value. The operating movement normally provided in hand or relay-operated electric switches for breaking heating load of, say, 2 to 3 Kw., can be obtained only at the expense of a reduction in operating force.

The development of the "Satchwell" or micro-gap switch just over ten years ago brought to the temperature-control field a method of switching which is ideal for thermostatic work, and in the temperature-control field is the most widely employed means of switching A.C. heating loads up to 3 Kw. The micro-gap switch depends for its operation upon the use of heavy contacts of high electrical and heat conductivity which are separated to a small distance only (of the order of 0.005 inch.) The small amount of heat generated by the short arc formed when the contacts separate is conducted away so quickly from the point of striking of the arc that with A.C. the contacts are sufficiently cool at the instant the current reaches zero in the A.C. wave, that the current does not persist after this point.

Only small amplification of the movement available from the direct expansion of a brass tube is required to provide the necessary movement of the contacts. This amplification can be obtained by the flexing of a spring lever without resource to pivots. Pivots are undesirable in thermostat design because they introduce points of wear, and the movement available for a given change of temperature is usually so small that wear is fatal to the continued accuracy of calibration of the instrument.

When the sensitive element comprises a piece of bi-metallic strip, one of the contacts may usually be mounted direct on to the flexible strip, so providing an exceedingly simple and robust arrangement, particularly suitable for room temperature control.

Another form of switch, employed to a lesser extent, is the mercury tube type, which has certain advantages when handling electric heating loads on D.C. supply. Amplification of the movement obtained from a bi-metallic system is usually necessary, and involves the disadvantage of lever and pivot arrangements mentioned earlier.

(2) The mechanical work obtained from the sensitive element may be employed to operate a valve on a gas, hot water, or steam supply line directly. In the construction of such a thermostatic valve, the vapor pressure or liquid expansion form of sensitive element is used in practically all except the smallest form of gas controls. The trapped vapor resulting from the heating of the volatile liquid in the bulb exerts pressure in the bulb, this pressure corresponding exactly to the temperature. A capillary tube transmits the pressure to the bellows which is connected directly to the valve, movement of which is opposed by a spring. Alteration of spring tension alters the temperature at which the vapor pressure overcomes the spring reaction and closes the valve. Such a device provides modulating control, since the valve takes up a throttling position dependent on temperature.

(3) For the purpose of providing modulating control by electrical means, the sensitive element may be used to move a contact over a slide wire resistance or potentiometer; here the sensitive element chosen is usually a filled system because of the large movement required by the slider. The slider resistance so operated is incorporated in an electric circuit of the Wheatstone bridge type, in such a way that a motorized valve is caused to take up a position corresponding with that of the slider on the resistance, *i.e.*, dependent upon temperature; alternatively, a set of motor-operated contacts may be operated in sequence to switch on more or less sections of an electric heating load according to temperature.

(4) Direct-acting thermostatic valves, similar in principle to those described earlier, may be used, not to control steam, gas, or hot water supplies directly, but as pilot valves providing increase or decrease of air or water pressure to a diaphragm, *i.e.*, pressure-operated valve of much larger size. The air or water pressure is piped to the diaphragm valve through a small orifice, and *via* the controlling instrument. At low temperature the thermostatically operated pilot valve allows full pressure to be applied to the diaphragm-operated valve, so opening it, while, as the temperature increases, the pilot valve directs more and more of the operating medium to a leak pipe or drain, so reducing the pressure on the diaphragm and allowing the valve to close.

(5) In a further development of the above system, the temperature sensitive element, a motor for providing oil pressure for valve operation, and the valve itself, are all built into one self-contained unit. The motor drives an oil pump which, when the thermostatic element demands heat, provides pressure on the large bellows which holds the main valve open. Increase in the temperature of the sensitive element causes this to open a small relief or pilot valve permitting release of oil from the large bellows chamber, and allowing the main valve to close. The position taken up by the valve varies exactly in accordance with the temperature.

Choice of Type of Instrument.

In considering what appears to be a large number of types of controls fulfilling similar purposes, it may be difficult to decide upon the merit or otherwise of a particular system, especiallv where the application is such that a number of different types will all fulfill the functions required. While it is not the purpose of this paper to make comparisons of the merit of different types as such, their broad characteristics render certain arrangements more suitable for some applications and so define the type of instrument to be used on any specific job.

Dealing with the various types in order:

(1) Bi-metallic-operated controls with short break switches are the direct means of controlling electric heating, particularly on A.C. circuits,

and they are therefore almost universally applied for the control of domestic electric water heaters and for the control of electric space heating. Owing to their inherent simplicity and robustness thermostats of similar type are widely used for all general purposes, *e.g.*, boiler and calorifier control, operation of motorized valves, &c.

(2) Bi-metallic instruments incorporating mercury tubes are advantageous where a low priced control is required for D.C. use, *e.g.*, for the control of D.C. electric water heaters. The reliability is not so great as that obtained with a bi-metallic thermostat of the micro-gap switch type used with a relay.

(3) Liquid expansion or vapor pressure instruments incorporating various forms of electric switch are useful where the space available for the sensitive element is small, or where it is impossible to accommodate the switch portion near the sensitive element. Typical examples are the control of small bakelite press platens, certain types of ovens, refrigerators, &c.

(4) Thermostatic valves of the direct-operation type are generally employed for small sizes of steam-heated tank and calorifiers; smaller types of directly operated valves, in which the expansion of a bi-metallic tube operates a valve directly, are used in domestic gas appliances such as cookers and water heaters.

(5) Electrical thermostats operating motorized valves are applied where the valve size or the pressure of the heating medium—steam, hot water, oil, or gas, as the case may be—demand the application of more power than is available from direct operation. When employed in this way a sensitive thermostat can command an electric motor capable of developing a high torque and operating valves or dampers of large sizes. Such a system is also invaluable when the sensitive element is located at a considerable distance from the valve to be controlled. Motorized valve operation, which can be either of the on-off, floating, or modulating type, can be considered the general purpose method of providing temperature control of steam or hot water heating systems.

(6) Air or water pressure-operated valves have the same field of application as modulating electric controls, and are advantageous where it is necessary to be independent of electric supply, and where air or water pressure is available. Such systems are often employed for the control of gas-heated boilers or ovens.

TRENDS IN STEAM PLANT DEVELOPMENT.

An abstract of an article by A. G. Christie, Professor of Mechanical Engineering, Johns Hopkins University, published in Combustion, for April, 1941. The author briefly reviews progress in the design of steam power-plant equipment to meet advances in steam conditions and operation, the influence on economy of generation and some indications of future trends judged by experience with present practice.

A prominent engineer in this country about 1910 stated in a discussion of the probable development of the steam turbine, that it was inconceivable that turbines would ever be needed of larger size than 25,000 Kw. Considering subsequent developments, how ludicrous his statement reads. This incident must impress one with the fact that the prediction of future power-plant developments is a dangerous undertaking. Caution must therefore be exercised in judging the trends herein discussed.

Several years ago the desirability and economics of the standardization of steam-plant equipment was seriously considered and some steps were taken to achieve this end. But developments in the power industry have not ceased and at present there is little standardization in much of the principal equipment that goes into a large power house. Trends in steam-plant development, judged by recent practice, are influenced by the following factors:

(a) The increasing use of alloys capable of standing higher temperatures
(b) Higher temperatures which permit the use of higher pressures without reheating
(c) The design of larger turbines at 3600 RPM.
(d) Tailor-made boilers and auxiliary equipment
(e) The introduction of forced-circulation boilers
(f) Improved station economy as a result of these changes
(g) The construction of more stations of moderate size
(h) Comparatively stationary total costs of production.

A study of a Mollier diagram indicates that more energy is made available from an increase of 100 degrees F. in temperature than from 100 pounds gain in pressure in the higher pressure ranges. Hence, as high steam temperatures have been employed as available metals will stand. At the moment this temperature limit appears to be about 1000 F. which has been fixed by the properties of commercial chrome molybdenum steels. Many turbines have been and are being built for temperatures of 900 to 960 F.

Research has been undertaken to find new materials with satisfactory properties at still higher temperatures. The introduction of the gas turbine as a power-generating unit and a realization of the gains to be attained with high temperatures on this unit, has acted as an incentive in the search for materials suitable for these conditions. Alloys containing columbium are said to offer much promise at higher temperatures. Undoubtedly, still other alloys will be found suitable for this purpose. Materials permitting the use of 1200 F. or even higher temperatures, may be made available in the near future. Such materials could be used quite as well on steam turbines as on gas turbines. Hence, one may expect further increases in steam temperatures before long.

Superheater tubes present a difficult problem. The metals of these tubes are subject to a considerably higher temperature than the steam inside, hence require more heat-resistant alloys than the turbine or piping.

With an assumed limiting moisture content in the turbine exhaust of 10 to 12 per cent, higher initial steam temperatures permit the use of higher steam pressures. Steam pressures up to 2000 pounds per square inch are in use and a 2400-pound plant is nearing completion. When temperatures of the order of those stated in the preceding paragraph are possible, namely, 1200 F., pressures up to 2500 pounds may be employed without reheating. There does not seem to be much gain at still higher pressures and the employment of the reheat cycle for such higher pressures involves costly steam-generator design.

The combined higher-temperature, higher-pressure expansion results in a greater heat drop. Even after allowance for decreased efficiency of the higher pressure blading, there will be a decrease in heat consumption per kilowatt-hour output over present types. With a given leaving loss from the limiting length of blades in the last row of the turbine, there can be an increased capacity of the turbine frame at the same time as increased economy.

Some may question the ability of the first row of turbine blades to stand higher pressures at light loads. Much development work has been done on these blades. The action of such blading is now better understood and

designers can meet this problem with more assurance of its successful solution than was the case a few years ago.

One of the first Westinghouse turbines, a 400-Kw. unit built for Yale & Towne in 1901 operated at 3600 RPM. The first Allis-Chalmers unit in 1905, of 5500-Kw. capacity and one of the largest then built, operated at 750 RPM. Since these early days there has been a steady trend in design to higher revolutions per minute for large turbines.

At the present time, turbine designers limit the length of the last row of blades to about 20 inches on 3600-RPM. units, partly due to wide spacing at the blade tips. This blade length with a 29-inch vacuum and normal leaving losses suffices for a turbine of about 25,000-Kw. maximum capacity in a single exhaust and about 50,000 Kw. in a double-flow exhaust on 3600-RPM. machines. At lower vacuum or with somewhat higher leaving losses, larger capacity units can be furnished. The largest condensing turbine-generator under construction at 3600 RPM. is the 65,000-Kw. tandem-compound unit for the new Harbor Station of the City of Los Angeles.

Some of the first large turbine-generators at 3600 RPM. developed objectionable double-frequency vibration in the generator frame due to the effect of the two-pole revolving field. This vibration frequently was transmitted to other structures and led to noise having a disagreeably high pitch. The difficulty has been met by suspending the stationary winding and core on spring supports from the frame. The first General Electric unit of this type, of 25,000-Kw. capacity, is operating satisfactorily at the Westport Station, Baltimore. The first Westinghouse unit, of 65,000-Kw. employing a different type of support, operated smoothly on test. The transmission of this vibration to the outer frame of large units thus appears to have been reduced to acceptable limits.

Hydrogen cooling is becoming standard for units above 25,000 Kw. and results in improved generator efficiency. Successful means to seal glands, and to admit purging of generators have been developed. Hydrogen pressures up to 15 pounds per square inch may permit an increase of about 15 per cent in generator capacity over that with hydrogen at about atmospheric pressure.

The limiting size of a single-shaft turbine-generator set at 3600-RPM. is now fixed by the maximum capacity of the generator, which, at the moment, is 65,000 Kw. The development of steels permitting larger rotor diameters and improved means for securing rotor coils will permit increasing the generator size. Larger units than 65,000 Kw. are cross-compound.

While turbines of 3600 RPM. may have better efficiencies than those of 1800 RPM. in sizes below 50,000 Kw., there appears to be little difference in efficiency in larger units due to high leaving losses. However, units at 3600 RPM. will be chosen when available in somewhat larger sizes due to their smaller physical dimensions and less liability to damage from temperature changes. At this time, there seems little justification for the purchase of 1800 RPM. units in sizes below 75,000 Kw.

Modern condensers are designed to secure such longitudinal distribution of the steam that all surfaces receive steam in proportion to their condensing capacities. At the same time low pressure drop across tube banks, maximum cooling of the uncondensable gases to the air pump and reheating to vacuum temperature in the hotwell are also desired objectives. Vertical types of circulating pumps either of axial flow or the adjustable-blade propeller class are being increasingly used due to space saving and other advantages, such, for instance, as placing the motor above flood stages. Further developments in condenser tubes are desired to increase their service life.

Many different steam-generator designs are available. One definite trend in all designs is toward the greater utilization of steaming surfaces exposed to radiant heat. Another trend is to give more attention to ample downcomers

and to positive circulation circuits for the boiler water. In general, the furnace walls form the main radiant heat absorbing surface, but consideration must also be given to the radiant heat emitted from the hot gases as they pass through the so-called convection surfaces. Sufficient convection surface may be placed between furnace and superheater to control the gas temperature to the convection superheater. Other convection surfaces beyond the superheater, if used, may serve as downcomers for circulating water and as means to reduce gas temperatures to economizers. The majority of the new steam generators purchased in the last few years are of the bent-tube type.

Superheater headers have nipples welded in place in the shops, and are annealed before shipment. The tubes are welded to these nipples in the field. There are many who believe that a similar procedure should be adopted for drums, headers and tubes of the boiler proper, in order to do away with the difficult operation of rolling tubes in place on high-pressure boilers to avoid leaks which often develop at the joints. Despite certain apparently valid objections that have been advanced from the standpoint of handling and erecting very large drums having welded nipples without incurring damage to the nipples, it is possible that such practice may eventually prevail for high-pressure work. In fact it is already being applied in the case of one large very high-pressure boiler now under construction.

Many types of steam washers and purifiers are in use, such as cyclones, bubble washers, wetted wire mesh and eliminators. The fluctuation of water levels over a considerable range with sudden load changes may require modifications of certain of these designs.

The importance of control to secure constant superheat increases with higher steam temperatures. Various arrangements effect this control, such as by-pass dampers, combinations of convection and radiant superheater surfaces, divided furnaces with radiant superheaters, and desuperheating by means of tubes submerged in boiler water or otherwise. Control may also be effected by providing adjustable burners that may be deflected downward at heavy loads and upward at light loads. In some cases, control is secured with horizontal turbulent burners placed at different levels with the upper burners only used on light loads. An objective in many plants is to secure constant steam temperature over a wide range of load by means of such devices, for instance, from 40 per cent to full capacity.

With steam temperatures of 1000 F. or above, the alloy tubes of superheaters will be glowing at their outlet ends. This raises the question of dissociation of the steam at high temperatures, which phenomenon may eventually prove to be the limiting factor in the use of high superheat. While research studies are being made, little data are at present available upon dissociation.

Power relief valves have been introduced to lessen the use and erosion of safety valves with high-pressure high-temperature steam.

Economizer designs appear to approach standardization with continuous pipe loops throughout. Present practice seems to favor a maximum water temperature leaving the economizer 50 F. or more below boiler temperature rather than the earlier use of steaming economizers.

Boiler feedwater conditioning must be given much thought to insure proper operation of high-pressure boilers. It is generally believed that deaeration in addition to that which may occur in a surface condenser is necessary for the optimum removal of dissolved oxygen. Control of pH, of soluble solids, of sulphate-alkalinity ratios, of any residual oxygen, and of boiler water concentration are factors in satisfactory operation concerning which much can still be learned. Silica in feedwater still presents difficulties, although much progress has been made in the development of means for removing a portion of the silica. Deposits by the steam of solids

on turbine blades have been lessened by careful water conditioning and concentration control but still require further study, particularly with regard to deposits of silica.

Piping needed with increased pressures and temperatures must be of alloy steel and quite thick. The design of this piping to prevent undue thrusts from expansion offers many difficulties. These pipe lines are now welded throughout.

The regenerative type of air preheater seems to be gaining preference over plate-type preheaters due to greater ease of maintenance and repairs.

Water circulation in boilers continues to receive much study. The difficulty of maintaining natural circulation with very high boiler pressures naturally suggests forced circulation, which is widely used in Europe and one large high-pressure unit is under construction here. Among the advantages claimed are: lighter weight through the use of tubes of smaller diameter; any desired tube arrangement may be used, as the water is pumped through all tubes; and the tubes are comparatively free from scale troubles. The gland difficulties of the circulating pump for this service appear to have been overcome. An increased use of such boilers may be expected.

Obviously, increased pressures and temperatures with constant vacuum increases the heat drop and thus decreases the station steam rate. Extraction heating with three to five bleed points, further improves station economy. A decrease in Btu. per kilowatt-hour should occur in these new high-pressure high-temperature stations.

A power station over its useful life will not operate continuously at either the most economical or the maximum load. Light loads must prevail during many hours of operation. Overall station performance at various capacity factors may be approximated by a form of Willans line. The slope of this line depends on the fuel requirements of the plant at no load with the main units spinning and at full voltage. The lower this no-load fuel consumption, the better will be the station economy at all loads, but particularly at light loads. A search for higher economy warrants a careful study of all auxiliary power requirements under such conditions in order to better the station performance as a whole.

The superpower plant may have some disadvantages in case of war. While this country is not likely to be subject to aerial bombardment, the merits of a number of smaller plants scattered over the area of distribution deserves careful study. The unit installation of one boiler, one turbine in sizes up to 80,000 Kw., together with the small labor force required, diminishes the advantages of the large plant. Outages on the whole system may be lessened by having several sources of supply available. Distribution losses may also be decreased.

A further factor to be considered in station size is chimney gases. Great volumes of rising hot gases are objectionable to airplanes. Since these heavy gases settle on cooling at varying distances from the chimney the concentration in surrounding neighborhoods must increase with increasing plant size. These factors favor a number of smaller stations.

Another consideration in chimney design is the objection from an aeronautical standpoint to tall chimneys. From a military point of view these offer excellent landmarks and range finders. Some German plants and several American stations have chimneys that do not project above the roof parapet. Gases from such chimneys tend to be carried down by eddies over the immediate plant surroundings. Hence a compromise on a medium chimney height appears desirable. Higher gas velocities may be needed at the chimney top to project the gas stream to a sufficient height to get the desired dispersion over adjacent regions.

PRELIMINARY CALCULATIONS IN MERCHANT SHIP DESIGN.

This is an abstract of a paper by E. E. Bustard, presented before a General Meeting of the North-east Coast Institution of Engineers and Shipbuilders, held in Newcastle upon Tyne on 28 February, 1941.

Preface.

In making the first approach to a new design, it is in all cases necessary to make certain approximations, in order to create a suitable foundation for more advanced calculations. The essential aids in preliminary calculations are reliable records from similar ships previously constructed. The value of such records and the accurary of comparative calculations are not generally realized, and preference is often given to rules and formulae derived from inconsistent data.

All designs have certain limitations imposed in the form of shipowners' requirements. Whilst satisfying these requirements as far as possible, and complying with the rules of the various governing bodies, the naval architect has to produce a design which will have a longitudinal balance, and be of proportions and fullness such as will ensure good sea qualities and driving properties. Strength, subdivision, and other subsidiary requirements are usually formulated for him by classification societies and international regulations, and he has only to see that the design satisfies their requirements. In certain specialized designs, however, he must decide upon suitable standards himself. The purpose of this paper is not an exhaustive study of the subject, but to put on record certain methods of dealing with various problems, which have proved useful in design. For the most part, it deals with merchant ships of conventional types, but many of the principles may be applied to specialized types provided that records of similar types are available. It should be understood that the paper has been intended to show means of arriving at a desired result, and does not pretend to put forward definite qualities to be aimed at in a design.

Dimensions and Form.

The influence of proportions upon the power required to carry a given displacement at a given speed is negligible within a wide range if suitable fineness and distribution of displacement are allied to the appropriate dimensions. This fact is used to advantage in design, in that dimensions are chosen to satisfy other conditions primarily, and a form designed suitable to those dimensions at a later stage. The conditions principally concerning dimensions are therefore the economic requirements of the merchant shipowner, which normally comprise: a deadweight carrying capacity on a stipulated draught; cubic capacity for cargo or equivalent deck area for passenger accommodation; and speed on service and radius of action; all in conjunction with a general arrangement suited to the ship's trade. The task of the ship designer is to satisfy these requirements, in so far as they do not conflict with one another, in the most economical manner consistent with seaworthiness and strength, to the advantage of both the shipbuilder and the prospective owner. An important consideration is that of weight, both because of its bearing on first cost and its subsequent influence upon operational charges. It is well known that length is an undesirable feature from the weight aspect, and the modern tendency is towards ships broader in relation to the length, allied with finer lines.

In determining dimensions for a particular design at the outset, one would be guided by a known ship similar in type to the proposed vessel. An estimate of all weights making up the displacement would be made for the provisional dimensions chosen, and if the summation of these did not check,

proportionate adjustment would follow. If no previous type ship or design could be found as a basis, the problem would be the same in principle, but would necessitate more labor and possibly a process of successive approximations. A deadweight-displacement ratio in conjunction with fixed proportions for the linear dimensions is sometimes used as a means of evaluating suitable dimensions; but this method is only of use in simple cases when no difficulty is experienced. Choice of propelling machinery will have a big influence, and specification differences are all against generalizations in ship design, even in ships of a similar type. The first indication of the form in ship design is generally given in the choice of a suitable block coefficient for the speed requirements. Model experiment results where available will prove the most reliable guide. Failing such information the well-known expression $C_B = 1.08 - \frac{V}{2\sqrt{L}}$ (V being the trial speed in knots) may be used with discretion for ships of normal proportions. A constant of about 1.04 in association with the speed on service may be alternately used in the above expression.

Where required for the development of the design, lines may be derived from a good basis form, lifting sections for the new design from the basis ship's lines at a revised spacing from the ends. This spacing is made inversely proportional to the differences between the relative prismatic coefficients and unity. Transverse dimensions may be altered by any convenient proportional means. The method is fairly well known, and it is sufficient to show a standard calculation here. (See Appendix 1.) Some notes upon experience gained in the use of the method are, however, of interest. First, if the longitudinal center of buoyancy is to remain in the same position relatively as that in the basis ship, then it should be seen that the relative displacement of the fore and after bodies is the same in the desired form as in the basis. This will insure that the relative moment of the form about amidships will not change to any noticeable extent. For this reason, it is better, particularly in very full vessels having the center of buoyancy well forward of amidships, to treat each body separately. Secondly, if it is required to move the center of buoyancy, it is necessary to know how to alter the relative displacement of the fore and after bodies so that the longitudinal center of buoyancy may move the required amount. In this case it is essential that each body should be considered separately. Calculations for a number of ships have shown that .02 prismatic coefficient removed from one body and added to the other, will move the center of buoyancy about .875 per cent of the length. In very full ships such as lake steamers with a block coefficient of about .85, this value would be about .90 per cent of the length, and correspondingly rather less in very fine vessels. The great advantage of the method is that the designer has initially full control over the ultimate result, both in regard to displacement and distribution of displacement as indicated by the position of the longitudinal center of buoyancy; and the body sections may be derived directly without fairing on the lines plan. Except for checking purposes, there is no necessity to make a displacement calculation for the derived form, as all particulars can be obtained from simple relationships with the basis form. These relationships are given in Appendix 2.

Lightweight and Deadweight.

After tentatively deciding upon suitable dimensions and block coefficient, the permissible draught will be fixed by law in conjunction with the general arrangement of the design. The load displacement, which must be disposed between hull and engines and the various items making up the deadweight,

is thus provisionally fixed pending confirmation of the chosen dimensions. The main problem remaining is the determination of the lightweight of the ship. Owing to the many variable factors involved, an exact estimate would be fortuitous, even if carried out with meticulous care. Rolling margin and scrap allowances in steel, and other weights, vary from ship to ship and cannot be foretold with exactitude. A suitable margin must therefore always be included in the lightweight to meet such contingencies. The lightweight itself is best estimated relatively from a previous ship, as by such a method various items difficult to estimate are automatically allowed for, and the time spent upon the investigation is considerably shortened. The same observations may be applied to center of gravity calculations for the light ship, as will be noted hereafter. Records of weights of previous ships, supplemented by data giving the changes in steel weight per foot change in each of the principal dimensions, and for rate of change with block coefficient, are of great importance for quick and reliable estimating in connection with new work when using the above method. Some difference exists in practice in regard to the tabulation of such data and subdivision of the weights into groups. One practice is indicated in Table 1, which also gives the rate of change in steel weight per foot change in each of the principal dimensions for a number of merchant ships. For changes in fineness, an approximate rule is to allow one-half to two-thirds of 1 per cent per .01 change in block coefficient. It should be noted that the changes given are without regard to scantlings, and it is usual to make an allowance of from one-quarter to one-third of the total difference in weight from the basis, to cater for such weight variation. A typical example of such a calculation is given in Appendix 3. In special cases, or when no suitable basis is available, more or less detailed calculations are entailed, presenting considerably more labor, as well as being subject to the disadvantage referred to in regard to unknown weights.

Speed and Power.

Speed being in the majority of cases a design requirement, estimates of the power required are of primary importance at the commencement. The number of published methods of estimating power is legion, but all are comparative methods based on previous experience, supplemented in many cases by data from model experiments. It is doubtful whether it would be an advantage to be able to calculate with any exactitude the power required to drive a ship at a given speed under certain assumed conditions, as the ship in service experiences a combination of factors seriously affecting resistance, over which no control may be exercised. These factors, such as fouling, weather and wind, not to speak of mechanical efficiency, can only be taken into account by statistical means; and therefore the estimate of power required to drive a ship at a given speed in service must always remain an approximation, and that with qualifications.

In recent years, model experiments have become so general in connection with new ships that a large amount of data is now available from which the EHP. of a design may be estimated in many cases by comparative methods from the results of such previous tests, taking into account differences in form and proportions. A statement of typical results converted to a standard length of 400 feet is given in Table 2 for a number of single-screw forms. Suitable increments for appendages and wind are indicated in Table 3. The BHP. or SHP. may be estimated by taking a suitable quasi-propulsive coefficient from previous tests or full-scale analysis, consideration being given to propeller revolutions per minute and the nature of the stern appendages. An allowance of from 3 per cent to 5 per cent for shafting losses, depending upon whether the machinery is aft or amidships, is included, this giving the power equivalent to smooth-water-trial

or tank-experiment conditions. A further increment of 15 to 25 per cent, according to size, fullness, and service, must be included to take into account average conditions at sea; and it should be borne in mind that in good practice the service power utilized is kept well within the maximum power which the engines are capable of developing under trial conditions. Suitable ratios of service power to trial power are .83 to .85 for steam machinery with coal-fired boilers; .85 to .88 for steam machinery with oil-fired boilers; and .90 for oil engines.

Where suitable bases are not available, the EHP. may be estimated from the "standard series" curves given in the "Speed and Power of Ships," by D. W. Taylor. In place of the diagram relating to frictional resistance in his presentation, Tables 4 and 5 may be substituted. These are derived from the same information as the diagram, and will be found easier and more consistent in use. The "standard series" of Taylor does not embrace the slow-speed cargo ships, and methods have been published by Baker, Ayre, and others which are useful in this respect, that of Ayre being particularly convenient. A simple table of Admiralty coefficients as given in Table 6 presents some advantages for practical use, with sufficient accuracy in the early design stages. In its first form it was prepared from the published results of model experiments by Baker, Kent, Robertson, Schaffran and others, but has been revised for the purpose of this paper to bring it more into line with modern forms. In using such approximate methods, suitable corrective factors derived from comparisons with actual results should be introduced, thus ensuring that the results are in substantial agreement with existing experience.

Where the revolutions in a new design differ widely from a given basis, corrective factors based on standard series of propeller experiments form a useful guide in assessing a suitable adjustment to the power. For convenience, Table 8 shows suitable corrections to power covering the range of average merchant ship practice. Figure 1 will also be found useful in this respect, covering cases outside those included in the table, and may also be of interest in considering other problems related to propulsion. It should be remembered, in using these factors, that they take no account of any modification of hull form to suit the propeller. If this is contemplated, they will be inclined to exaggerate the effect of change. Again, where blade widths are increased to defer cavitation, a suitable reduction in propeller efficiency should be allowed, if not already covered by the comparison with the basis form. In connection with propeller problems it should be remembered, when studying the results of model experiments, that the wake factor or wake fraction derived from the model is materially greater than would occur in the full-scale ship. For preliminary estimates the average wake fractions given by Taylor for design purposes will be found useful in normal ships.

Capacity.

For all purely cargo ships, the cargo capacity can be rapidly approximated with considerable accuracy by a method involving the use of coefficients taken from similar ships. To this end, it is useful to have tabulated the total moulded volume of all spaces above the tank-top and below the uppermost continuous deck for previous ships, and to express this volume as a ratio to the product of the length between perpendiculars, breadth moulded and depth of hold. It is also convenient to think of this ratio as an excess over the block coefficient of displacement, and thus eliminate variations due to fineness. Care must be taken to compare fineness at draughts the same proportion of the depth moulded in the basis and desired cases. Deduction of the non-cargo spaces in the design presents no difficulty, and differences in general arrangement, sheer and erections can readily be allowed

for. Table 9 gives typical data for a number of ships, and a sample calculation is shown in Appendix 4.

In cargo ships, it is common practice to state cargo capacity under two conditions. First, grain measurement in which deduction from the moulded volume of the actual volume of any obstructions only is made. This may be taken at about 1½ per cent less than the moulded volume. Secondly, bale measurement which is normally taken to the inside of frames or cargo battens, bottom of overhead beams, and top of tank-top ceiling or deck. Under the above conditions, the bale measurement is usually less than the grain measurement by about 10 per cent in cargo ships. It is advisable, however, to use some past ship as a guide in assessing a value for the deduction to be made in any particular case, having regard to the height of the cargo spaces, number of decks and any other important differences.

The more elaborate modern refrigerated cargo ships, particularly those engaged in the meat trade, cannot normally be dealt with by methods involving the consideration of the whole ship as one entity. Each compartment must be treated on its merits, because of the proximity to one another of spaces falling into widely different categories. Having ascertained the moulded volume of the various spaces, however, the deductions for each can usually be assessed on a percentage basis, having regard to the size and height of the space, and to the nature of the refrigeration adopted. Typical figures are given in Table 10.

In estimating the cargo capacity of tank ships, it is very convenient to have tabulated the moulded volume of the tank range for a number of type ships, including all intermediate pump rooms and cofferdams. This volume may be expressed as a ratio to the product of the length of the range and the 'midship area. Pump rooms and intermediate cofferdams can be deducted on their merits in utilizing such information in connection with a new design. Some typcal figures are given in Table 11, which also gives the center of gravity of the volume of the tank range in relation to the middle of the range. In ships having normal sheer, a useful approximation to the center of gravity of the tank range is one foot forward of the middle of the range for normal-sized tankers.

Tonnage.

It is perhaps unfortunate that such a purely incidental thing as tonnage should place any restriction upon the free development of a design. Nevertheless, due to imperfections of the Tonnage Act, it is so. In the initial stages of design, it is necessary to satisfy the condition that the engine-room tonnage shall be not less than 13 per cent of the gross tonnage. The underdeck tonnage can be rapidly approximated on a coefficient basis from a similar ship. The coefficient is taken as follows:—

$$\frac{\text{Underdeck tonnage} \times 100}{\text{Length B.P.} \times \text{Breadth moulded} \times \text{Tonnage depth.}}$$

The excess of this coefficient over the block coefficient may be tabulated for various ships and utilized in a similar manner to the capactiy coefficient as previously described. In determining the gross tonnage, the additions for erections can be determined by measurement from the outline plans.

Trim.

While trim calculations are in principle simple, the results are of great practical importance. Upon the naval architect's judgment in allocation of weights and distribution of displacement, the attainment of a good balance making for ease of operation depends. There are two main aspects: the

balance of the ship under one main condition of loading, which brings in the position of the longitudinal center of buoyancy as an important factor; and, the relative balance between the main and other conditions due to consumption of fuel, etc., or, alternative conditions of loading. The basis for these calculations, as far as the cargo is concerned, is usually given by the shipowner based upon his knowledge of average cargoes experienced in the desired trade. The consumable weights depend on the requirements of the propelling and auxiliary machinery, together with the number of passengers and crew in association with a specified radius of action. A suitable allocation of weights for this part is usually the naval architect's responsibility, working in collaboration with the marine engineer and his needs. The primary problem in trim calculations, as in weight and stability estimates, is the position of the longitudinal center of gravity of the light ship; and similar considerations apply, the relative method of calculation being particularly desirable owing to the considerable effect a discrepancy in the end weights may have in a detailed calculation. In the case of ships having machinery aft and designed for making long ocean voyages, it is advisable to have approximately half the oil fuel disposed in the forward end, and to provide a margin of capacity above the minimum requirements at each end, in order to have some latitude in trimming. This would provide a margin in case the position of the longitudinal center of gravity did not come out quite as estimated, and would also make the ship handier from the point of view of balance in service. In large vessels, the condition for dry-docking may be of some importance, requiring consideration in preliminary design.

A useful approximation to remember in connection with trim calculations, is that the longitudinal GM and longitudinal BM are for all practical purposes very nearly equal to one another, and therefore the trim in feet can be equated to the product of the lever causing trim and the ratio length over longitudinal BM. This ratio varies somewhat with the draught, but this can be taken care of by selecting a value from a similar ship at the appropriate draught. Typical values of the length-LBM. ratio are included in Table 1. It will be noted that a value of about .83 represents a good average for this ratio at the load draught, and therefore ten times the lever causing trim will represent the trim in inches, with fair approximation at this draught. Since one is not usually concerned with abnormal trim in design, any error involved in this assumption is of no practical importance.

Stability.

As already stated, it is not the purpose of this paper to discuss standards to be aimed at, but to describe briefly means of arriving at a desired result. In the first place, the designer would choose tentative transverse dimensions and form suitable to the arrangement and loading of the design. Reference would be made to data from previous ships, with due regard to the extent of superstructures. The metacentric height would then be estimated for the dimensions provisionally chosen for some basic loading, and subsequent adjustment made to insure that the desired result might be attained. The only factor that need be to some extent doubtful in the design stages is the vertical center of gravity of the light ship. For the hull part, this is best estimated by comparative methods from a previous ship. Where no data are available from a similar vessel, recourse may have to be made to a detailed calculation. While this may seem to be the best method from the academic standpoint, it is open to a number of serious objections in practice. In addition to the number of small errors involved in the summation of a large number of items, there is always the possibility of a number of items being overlooked, while many are in themselves difficult to estimate. In making a comparative calculation, small items particularly liable to omission,

and the multifarious items which are doubtful, are automatically included on a known basis. A typical relative calculation which shows clearly the method is given in abbreviated form in Appendix 5.

The designer usually has in mind in the early stages some typical basis form, and the height of metacenter can be conveniently estimated for the design therefrom by coefficients, the vertical center of buoyancy above the basis being considered as a fraction of the draught, and the height of metacenter above the center of buoyancy as a ratio of the beam squared to the draught. Values of these coefficients are given in Table 1 for a number of forms. If the lines are available, the metacenter may be derived directly from the form, eliminating any uncertainty in this respect. It should be remembered that, in twin-screw designs, the after waterline can often be filled out considerably without appreciable detriment to the performance, and an increase in transverse dimensions avoided in certain cases, where an increase in metacentric height appears to be required.

A condition often included in the design requirements as formulated by shipowners or their representatives, is that of a certain metacentric height in light condition. This is a very unsatisfactory criterion in the majority of designs, as the curve of metacenters rises steeply towards the lighter draughts, whereas it varies but little over a wide range near the load draught. It follows that any error in the estimate of lightweight can cause a considerable variation in the metacentric height in the light condition without appreciably affecting the results in the loaded conditions; and an exact estimate of the final lightweight of a passenger ship in the design stages would be purely good fortune. Reference should also be made to the notes on longitudinal subdivision and its bearing on the question of stability.

Subdivision.

In the initial stages of passenger ship design, the question of watertight subdivision has a great bearing on the lay-out of the passenger spaces, passenger services, and machinery spaces; particularly in large high-speed vessels. The so-called "criterion numeral" will also be required in order to cover for the requirements of international law. A number of elaborate methods have been put forward for finding the "curve of floodable lengths" from the geometrical form of the hull. In view of the fact that the standards of subdivision required are of necessity based on a number of arbitrary factors—such as permeabilities of the various spaces, factors of subdivision, extent of assumed damage, etc.—and that the standard of safety attained is purely relative and not absolute, it would appear that extreme accuracy in the method of calculation is not essential. The approximate method put forward by the "Bulkhead Committee" being legally recognized by international law, gives all the accuracy required.

The "permissible length curve" for the "standard form" may be readily determined in the early stages of the design; and there may be critical points where this does not give complete satisfaction, and more exact information may be desirable. The "correction for form" involved in the use of this method is troublesome in the first approach, when in any event ideas about form are usually either in a state of flux or have not been given practical consideration. For guidance in such cases, tabulated corrections for form for a number of previous ships form a valuable aid. Typical figures are given in Table 12. In applying such corrections, it should be remembered that the shape and fullness of the mean waterplane, together with the distribution of the total volume, are the chief factors affecting them; and due consideration should be given in the choice of a suitable ship as a basis. For example, the corrections stated for a ship with an elliptical stern should on no account be applied to a design having a cruiser

stern. As a first approximation, the permeabilities of the fore and after ends may be derived from formula:—

$$\mu = 64 + 36\frac{h}{H}$$

where h = mean depth of 95 per cent permeability spaces below the margin line, excluding the double bottom, over the length under consideration.

H = depth of ship below margin line at half-length of part considered.

Later, it is not a difficult matter to estimate these with a greater degree of accuracy from the plans. The whole volume below the margin line for these spaces presents some complication, and a detailed calculation can usually be avoided by estimating the total volume below the margin line for the whole ship, in a similar manner to that previously described for cubic capacity. This volume may then be split up into fore and after ends and machinery space, broadly on the basis of some previous ship, making due allowance for differences in the lengths of the spaces. The permeabilities are not sensitive to extreme accuracy in these particular volumes, and this procedure will be found sufficiently accurate in the design stages. The necessary particulars to enable such a calculation to be made are included in Table 12 for a number of ships.

In the approach to direct flooding there has been a tendency to attack the subject from an unpractical angle, with resulting complications. In the majority of cases the exact floodable length is not required. It is generally only necessary to determine whether or not a given compartment satisfies the conditions; and it may be noted that in the design stages it may not be necessary to consider all compartments. Investigation of the self-evident critical regions will often suffice. A simple trim calculation, assuming the compartment in question completely filled at the rule permeability, will often determine that the margin line would not be immersed; in which case more exact calculations are redundant.

Longitudinal subdivision is a possible source of danger in the event of side damage. This may require the provision of what may be considered an excessive metacentric height in the fully-loaded condition, in order that the ship may have an adequate margin of stability in the least stable service condition to remain in a safe condition after sustaining damage. This applies to practically all large passenger ships and to all warships of the cruiser and capital-ship classes. The requirements as regards possible damage are of necessity much more severe for a warship than for a passenger vessel, and these must be dictated by the amount of damage expected as the result of accident or, in the case of the warship, sustained enemy action. The damage to be assumed in the case of passenger ships to meet hazards of service is stipulated in the Board of Trade "Instructions as to the Survey of Passenger Ships." The determination of the angle of heel with wing compartments laid open to the sea on one side is troublesome in the design stages, as it is difficult to forecast the final outcome of important factors affecting it. Cases differ materially, and each must be considered on its merits. Within certain limits the provision of automatic levelling arrangements provides means of adjusting matters. It is advisable, however, that such arrangements should not adversely affect the stability of the ship in any condition, damaged or undamaged. An arrangement of cross-connecting pipes fitted in a recent large liner for this purpose is illustrated in Figure 2. Alternative arrangements which would permit cross flooding at an earlier stage and thus reduce the initial heel after damage are shown in Figure 3. It should be noted that the question of stability when damaged may enter into the ordinary straightforward case of plane

transverse bulkheads, without longitudinal subdivision. It does not seem rational that the requirements of international law should ignore this point and yet show great interest in it as soon as a case of longitudinal subdivision arises. Admittedly, the latter case is the more dangerous, but in certain cases the other may be serious also.

Strength.

In the case of all seagoing ships, the scantlings and structural arrangements are the outcome of previous experience, the actual stresses developed at sea being mainly indeterminate. Such experience is embodied in the rules of the various classification societies, from which the scantlings of a new design are normally accessible. Occasionally, however, it is necessary to determine scantlings for a design which is not covered by classification rules, and which is either to be built to some classification society's requirements, or, in cases such as warships, is to have strength characteristics similar to an existing ship which has been found satisfactory in service. In deciding upon suitable scantlings, the longitudinal strength in a static condition upon a standard wave is often considered, the strength members being made relatively equivalent either to some orthodox type covered by classification rules, or to some ship of similar construction which has proved satisfactory in its strength qualities. In making such calculations it is usual to base scantlings upon the maximum bending moment. It may be pointed out that, in ships where the loading is clearly defined, there is no need to carry out calculations covering the full length of the ship. As the maximum bending moment will in all cases occur in the vicinity of amidships, it is convenient to calculate the moment about amidships. The maximum bending moment may then be found by making the small correction to the point of zero shearing force. By this method it is often possible to dispense with the usual load curve and the labor involved in its preparation. The subsequent operations, moreover, are rendered much simpler and more precise. A specimen calculation given in Appendix 6 shows the method in detail.

Vibration.

In recent years, detailed vibration calculations have been developed to a considerable degree, chiefly in connection with the primary vertical frequency, but it is doubtful if they can be applied to ships of elaborate construction, or to higher modes than the fundamental. Moreover, integral calculations require a rather more complete knowledge of the ship than would normally be available in the early design stages. In view of this, and as a certain amount of variation in the frequency may occur according to the distribution of the loading, apart from that due to variations in draught in service, approximate formulae will suffice for the degree of accuracy possible. The primary vertical frequency may be estimated from the approximate formulae proposed by Schlick, Dr. Brown, L. C. Burrill, and others, using constants derived from observations made on trial trips, supplemented by other records. The Schlick formula is not so reliable as the others if one applies a particular constant to obtain results over a wide range of draughts in any particular case. Experience and accumulated data will, of course, enable one to avoid this source of error.

A little uncertainty exists with regard to an appropriate formula for the horizontal case. So long as one is only concerned with figures at the load draught, no appreciable error will be involved if the effect of entrained water is ignored. Under these conditions experience has shown that a simple Schlick expression will give results as consistent as any more complicated formula. When considering draughts other than the load, however, a factor taking into account the entrained water is necessary to bring the

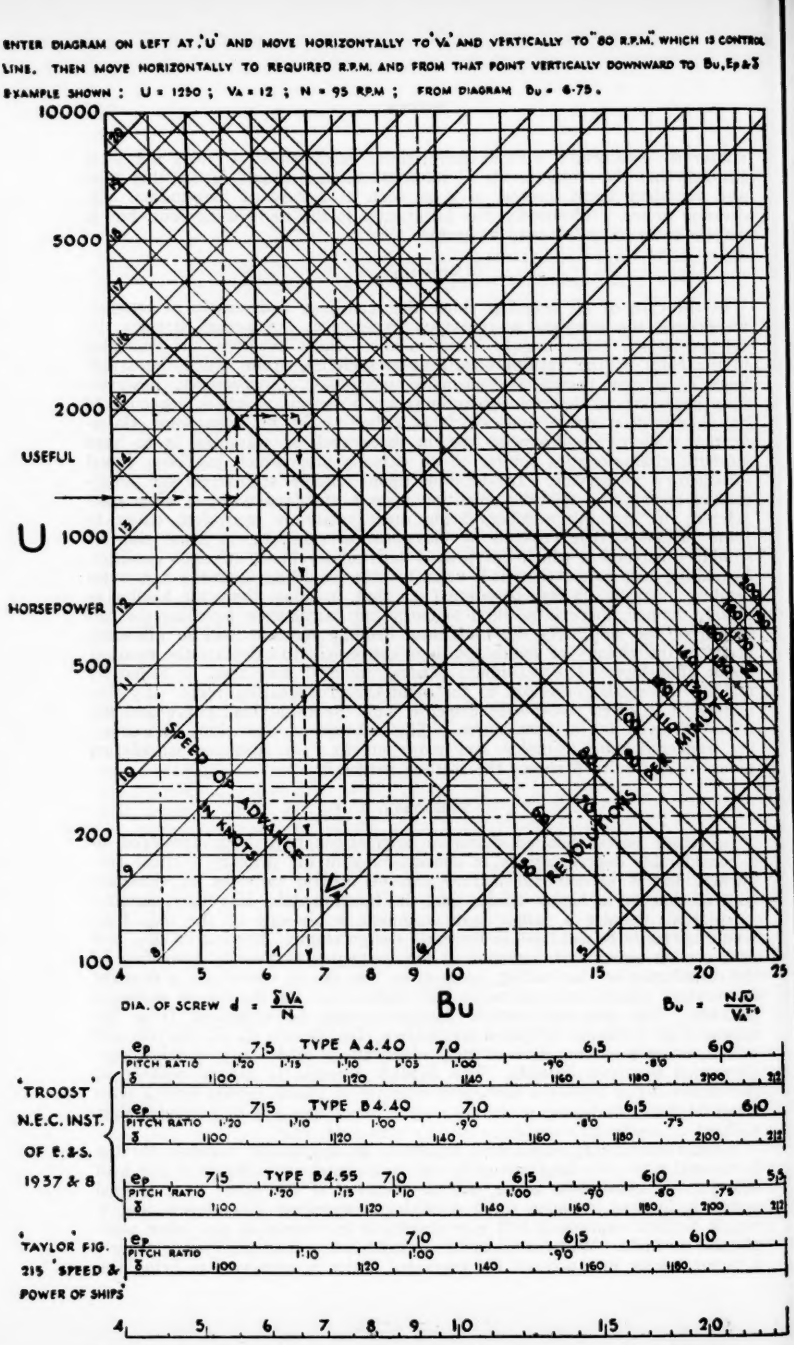

ENTER DIAGRAM ON LEFT AT 'U' AND MOVE HORIZONTALLY TO 'VA' AND VERTICALLY TO "80 R.P.M." WHICH IS CONTROL LINE. THEN MOVE HORIZONTALLY TO REQUIRED R.P.M. AND FROM THAT POINT VERTICALLY DOWNWARD TO Bu, Ep & δ
EXAMPLE SHOWN : U = 1250 ; VA = 12 ; N = 95 R.P.M ; FROM DIAGRAM Bu = 6·75 .
10000
5000
2000
USEFUL
U 1000
HORSEPOWER
500
200
100
SPEED OF ADVANCE IN KNOTS VA
REVOLUTIONS PER MINUTE
4 5 6 7 8 9 10 15 20 25
DIA. OF SCREW d = δ VA / N
Bu
Bu = N√U / VA^2.5
'TROOST' N.E.C. INST. OF E.&S. 1937 & 8
'TAYLOR FIG. 215 'SPEED & POWER OF SHIPS'
Ep
PITCH RATIO
δ
TYPE A 4.40
TYPE B 4.40
TYPE B 4.55
4 5 6 7 8 9 10 15 20

Fig. 1. *Alignment Chart for Optimum Propellers based on Results of Open Water Tests.*

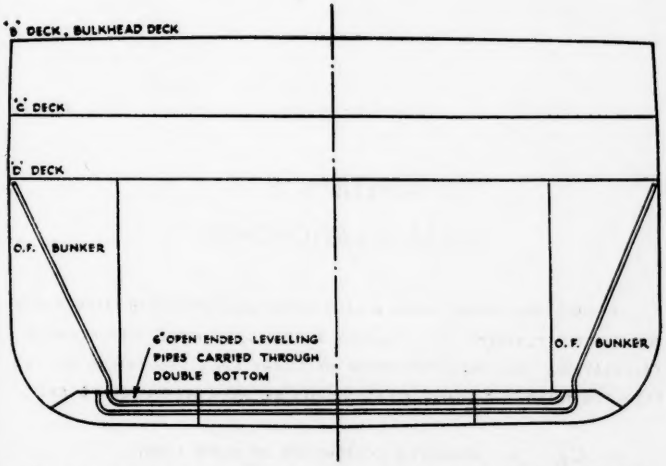


Fig. 2. Automatic Levelling Arrangements to Wing Bunkers fitted in Recent Large Liner.

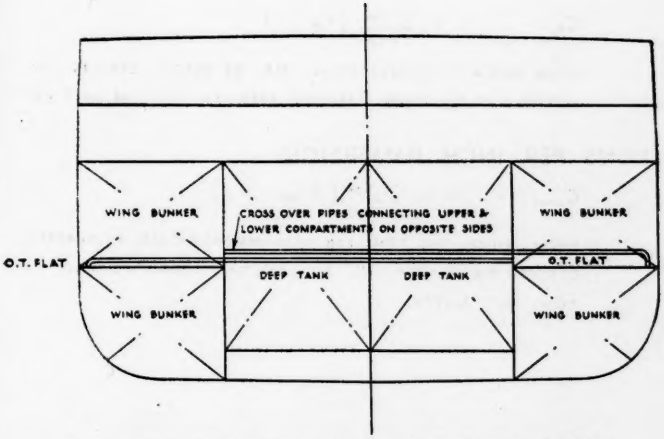


Fig. 3. Suggested Alternative Arrangements for Counteracting Heel in event of Side Damage.

APPENDIX 2.

FORM RELATIONSHIPS

IF LINES ARE DERIVED FROM A BASIS FORM AS DESCRIBED IN PAPER, THERE IS NO NECESSITY, EXCEPT FOR CHECKING PURPOSES, TO MAKE DISPLACEMENT CALCULATIONS. ALL THE INFORMATION NECESSARY FOR DESIGN CAN BE DERIVED FROM THE FOLLOWING SIMPLE RELATIONSHIPS BETWEEN DESIGN FORM & BASIS.

C_{P_1} = PRISMATIC COEFFICIENT OF BASIS FORM
C_{P_2} = " " - DERIVED FORM
C_{WPA1} = WATERPLANE AREA COEFFICIENT FOR BASIS FORM
C_{WPA2} = " " " - DERIVED FORM

DISPLACEMENT

$$C_{P_2} = 1 + \frac{1 - C_{P_2}}{1 - C_{P_1}} \left(C_{P_1} - 1 \right)$$

FROM WHICH THE DISPLACEMENT MAY BE READILY DERIVED BY MULTIPLYING BY LENGTH × MIDSHIP AREA ÷ 35 FOR TONS SALT WATER.

TONS PER INCH IMMERSION

$$C_{WPA2} = 1 + \frac{1 - C_{P_2}}{1 - C_{P_1}} \left(C_{WPA1} - 1 \right)$$

FROM WHICH THE TONS PER INCH IMMERSION MAY BE READILY DERIVED BY MULTIPLYING BY LENGTH × BREADTH ÷ 420' FOR TONS SALT WATER

TABLE I : PARTICULARS OF WEIGH

Index letter		J	K	Q	R	S
Year of completion		1927	1930	1932	1932	1935
Type		lake cargo	fruit	passenger & cargo	cargo	oil tanke
Length B.P.		253'-0"	280'-0"	245'-0"	248'-0"	460'-
Breadth moulded		43'-2"	59'-10"	37'-6"	59'-4"	59'-
Depth moulded		20'-0"	25'-6"	21'-0"	24'-4"	34'-
Erections		sunk f'cle & raised quarter deck	f'cle & sunk poop	f'cle & long bridge	shelter deck	poop bridg & f'cl
Machinery : type		Steam T.E.	Steam T.E.	Steam T.E.	Steam T.E.	Mota
" : position		aft	amidships	amidships	amidships	aft
Number of propellers		1	1	1	1	1
Net steel & rivets	tons	675½	857½	711¾	710	315
Stem, stern & rudder frames	"	6½	12½	7	7	2
Solid pillars	"	1¾	25¼	11	3	
Smithwork	"	8½	18¾	8½	13	2
Castings	"	11¾	14¼	8¼	11	3
Cement	"	6¼	29¼	8¼	10	1
Cargo refrigeration	"	-	-	-	-	-
Wood & deck coverings	"	142¼	129	260¾	60	9
Outfit & remainder	"		168¾		98	39
Hull & outfit	tons	853	1255	1016	912	395
Machinery	"	161	434	255	177	59
Lightweight	tons	1014	1689	1271	1089	454
Deadweight	"	2616	2194	1621	2635	1219
Displacement	tons	3630	3883	2892	3724	1673
Load draught, top of keel		14'-0⅝"	18'-10"	13'-2⅝"	16'-7¼"	27'-
Block coefficient : B.P. length		·850	·645	·722	·800	·78
Weight per foot of length		2·25	2·45	2·11	2·26	5·3
" " " " breadth		9·13	11·17	9·44	9·23	26·3
" " " " depth		8·52	8·58	10·40	7·42	42·5
Long'l C.G of light ship abaft ✕		16·5'	7·82'	6·80'	2·50'	24·
Vertical C.G " " above T.K		13·03'	17·04'	16·59'	15·30'	22·5
V.C.B. / DRAUGHT	AT LOAD DRAUGHT	·515	·542	·530	·536	·52
	AT 75% LOAD DRAUGHT	·517	·543	·531	·540	·52
B.M. × DRAUGHT / BREADTH²	AT LOAD DRAUGHT	·0810	·0795	0811	·0806	·080
	AT 75% LOAD DRAUGHT	·0812	·0781	·0798	·0797	·078
LENGTH / L.B.M	AT LOAD DRAUGHT	·77	1·05	·95	·96	·81
	AT 75% LOAD DRAUGHT	·59	·86	·75	·78	·68

WEIGHTS ETC.. FOR TYPICAL SHIPS.

R	S	U	V	W	Y	a	b	d	f
32	1935	1935	1936	1937	1937	1937	1937	1939	1939
rgo	oil tanker	cargo	oil tanker	refrig. cargo	passenger & cargo	cargo	cargo	passenger & refrig. cargo	oil tanker
'- 0"	460'-0"	435'-0"	425'-0"	495'-6"	410'-0"	410'-0"	435'-0"	650'-0"	500'-0"
'-4"	59'-0"	55'-9½"	54'-3"	68'-0"	55'-9½"	57'-3"	58'-9½"	84'-6"	64'-3"
'-4"	34'-0"	32'-0"	31'-0"	41'-6"	31'-0"	38'-0"	39'-9½"	48'-6"	37'-0"
elter ck	poop, bridge & f'cle	poop, bridge & f'cle	poop, bridge & f'cle	shelter dk. and poop, bridge & fcle	poop, bridge & f'cle	shelter dk and forecastle	f'cle & shelter dk.	two tiers & large deckhouses	poop, bridge & f'cle
m T.E. dships	Motor aft	Steam T.E. amidships	Motor aft	Motor amidships	Steam Q.E. amidships	Motor amidships	Motor amidships	Motor amidships	Motor aft
1	1	1	1	2	1	1	1	4	1
10	3156½	2513	2614½	4267¼	2616¼	2102	2458	9790	4370
7	28¾	24	25¼	42½	25¾	24¼	25	110	29
5	8¼	7¼	9¼	8	6	8¼	5¾	30	10
13	21¼	20¾	19¾	37½	29	24¼	22¾	80	22
11	39¾	45	35¼	49¼	45½	19	32½	60	56
10	12	5 1¾	16½	61¼	16¼	25¼	24¾	125	20
-	-	-	-	1845	-	-	152¼	1915	-
60	93¼	157¾	80	295¾	1009¼	177	190¾	1550	117
98	595½	351½	418½	420½		251½	346¾	1800	554
12	3954	3179	3219	7047	3748	2652	3256	15220	5168
77	590	826	579	1515	1056	603	709	3640	793
89	4544	4005	3798	8560	4804	3255	3945	18860	5961
55	12190	9310	9185	10905	6940	9780	9920	17360	15265
24	16734	13315	12983	19265	11744	13015	13865	36220	21226
'-7¼"	27'-4"	25'-6¾"	25'-5⅞"	28'-3⅝"	25'-5½"	25'-10⅜"	26'-1⅛"	34'-0"	29'-5¼"
300	·786	·749	·771	·705	·703	·748	·727	·677	·782
·26	5·34	4·65	4·56	7·14	4·70	4·32	4·67	10·6	6·6
·23	26·35	23·26	24·73	32·60	23·81	18·65	22·95	58·7	34·6
·42	42·33	20·62	40·90	27·08	22·68	17·70	18·05	48·0	54·1
·50'	24·5'	7·60'	26·37'	9·71'	11·67'	·27'	8·68'	20·05'	27·85'
·30'	22·52'	21·60'	21·59'	28·29'	24·62'	21·48'	22·83'	37·04'	25·96'
536	·521	·531	·524	·543	·533	·5305	·531	·553	·521
540	·520	·531	·524	·544	·534	·529	·530	·555	·520
806	·0803	·0818	·0811	·0835	·0795	·0825	·0810	·0854	·0802
797	·0787	·0805	·0793	·0812	·0779	·0809	·0790	·0827	·0785
96	·81	·78	·82	·75	·88	·85	·81	·685	·80
78	·68	·66	·68	·63	·75	·72	·70	·555	·68

Illustrating " Preliminary Calcul

by E. E. Bust

TABLE 9 : CAPACITY DATA FOR CA

Index letter	G	h	H	i	
Length B.P.	400	237	406·5	455	2
Breadth moulded	53·25	35	52	58·17	3
Depth moulded	36·25	17·75	31	34·58	2
Designed load draught moulded	24·81	16·83	25·47	28·15	1
Block coefficient	·777	·750	·723	·747	·
Sheer :- forward	9·50	8·00	9·92	10·17	6
Sheer :- aft	4·75	4·00	5·19	5·74	3
After peak	16·0	11·50	20·00	18·00	1
Length of after holds	126·58	65·17	119·17	165·50	8
Engine room	46·50	38·33	86·66	57·50	4
Length of forward holds	189·92	109·25	156·00	188·50	12
Fore peak	21·00	12·75	24·67	27·50	1
Mld volume above tank top & below dk.	621,250	111,250	491,900	702,500	19
Depth of hold	33·75	15·75	28·46	32·08	2
Capacity coefficient = Moulded volume / L. × Br. × D. of hold	·866	·856	·816	·827	·
Excess over block coefficient	·089	·106	·093	·080	·
Midship area	1760	537·5	1432·2	1852	9
Mid. area ÷ (Br. mld × D. of hold)	·979	·976	·981	·982	·
Depth of tank	3·63	2·75	3·38	3·67	2
V.C.G. of cargo volume above tank top corrected for peaks, mach. space & tunnels	18·77	9·35	16·62	17·94	1
V.C.G. as a proportion of depth of hold above tank top.	·556	·594	·583	·559	·
L.C.B at designed draught	-2·90	- ·55	- 1·96	- ·94	+
Number of complete decks	2	1	2	2	3 F 2
Number of rows of pillars	1	1	1	2	1 & 2 Of
Percentage deduction from mld for bale	10·35	8·6	12·0	11·07	1
Remarks:	Ceiling over bilges and under hatches only: 2" sparring below second deck.	2½" ceiling over bilges and under hatches only: 2" sparring	2½" ceiling over bilges and under hatches only: 2" sparring	2½" ceiling over bilges and under hatches only: 2" sparring	front bale to m fram unde beam ceili deck

Note:- all dimensions, areas, and volumes, are in foot units. The + sig

FOR CARGO SHIPS

i	K	j	U	k	W	Y	b
455	280	486·5	435	410	495·5	410	435
58·17	39·85	65	55·79	57·25	68	55·79	58·79
34·58	25·50	35·25 45·86	32·0	38·0	35·42 41·94	31	39·46
28·15	18·85	29·71	25·56	25·91	28·30	25·46	26·07
·747	·649	·720	·749	·748	·705	·705	·7266
10·17	6·00	9·00	10·00	7·00	9·54	9·52	8·25
5·74	3·00	4·50	5·90	3·31	4·85	4·19	4·00
18·00	16·17	19·04	18·00	22·00	19·04	20·00	26·00
65·50	80·16	175·75	150·75	156·50	176·00	127·85	136·00
57·50	47·67	63·25	67·50	45·00	63·25	82·35	61·35
88·50	121·35	204·00	175·50	165·75	212·25	158·17	187·92
27·50	14·67	24·46	23·25	20·75	24·96	21·67	21·75
702,500	191,750	809,100 1,031,900	609.500	693.600	804,200 1.046,700	520,100	778,800
32·08	25·42	31·55 40·15	29·46	35·54	29·80 37·88	28·55	36·94
·827	·735	·811 ·828	·8525	·8359	·801 ·830	·796	·8282
·080	·088	·091 ·108	·1035	·0859	·096 ·119	·095	·1016
1852	912	2014 2566	1615	1998·2	1986 2529	1570·5	2155·8
·982	·978	·982 ·984	·985	·982	·980 ·982	·986	·985
3·67	2·92	5·00	3·67	3·65	5·00	3·58	3·71
17·94	13·35	17·75 23·25	16·77	19·25	16·80 21·06	16·35	20·27
·559	·570	·561 ·595	·570	·542	·564 ·586	·572	·548
−·94	+·80	+3·38	−8·26	−3·36	+3·56	−·87	−3·72
2	3 FORWARD 2 AFT	3	2	2	3	2	3
2	1 & 2 IN WAY OF HATCHES	2	2	3	2	2	2
11·07	17·7	Insulated	9·6	10·0	Insulated	11·3	10·2
' ceiling over bilges and der hatches only: sparring	(mail steamer: bale capacity to inside of frame lining, underside of beams, top of ceiling and deck ceilings.		2½" ceiling over bilges and under hatches only: 2" sparring below second deck.	2½" ceiling over bilges only. 2" sparring		2½" ceiling over bilges and under hatches only: 2" sparring.	2½ ceiling over bilges and double bottom tank top: 2½" sparring

The + sign denotes aft of amidships, and the − sign forward of amidships.

Illustrating "Preliminary

by E.

TABLE 12 : SUBDIVISION PARTICU

Index letter			A	B	C	D
Length B.P.			601·42	600·5	519	485
Subdivision length			631	600	518·21	500
Breadth moulded			76	75·5	65	59
Depth moulded			50·2	45·0	45	45
Depth to margin line			50·17	45·04	42·92	42·8
Sheer :-	forward	on m.l.	12·25	11·5	8·92	11·0
	aft	"	5·92	6·0	4·42	4·0
Subdivision draught			30·00	30·00	29·21	29·25
Block co. on subdivision length.			·652	·717	·732	·704
Total volume below m.l. "V"			1,799.600	1,587,000	1,174,600	1,006,50
Volume coeff. $\frac{V}{L \times B \times D \text{ to m.l.}}$			·748	·800	·814	·796
Excess over block coefficient.			·096	·083	·082	·092
Length of forward end			192·91	284·67	173·99	209·92
" " machinery space			228·36	138·66	98·66	81·50
" " after end			209·73	176·67	245·36	208·58
Volume of forward end			445,500	751,400	547,750	418,20
" " machinery space			838,050	452,300	272,300	202,10
" " after end			516,050	383,300	354,550	386,20
Permeability of fore end		%	82·6	73·2	74·3	67·4
" " mach. space		%	80	80	80	80
" " after end		%	86·6	80·4	77·2	68·9
Freeboard ratio			·672	·501	·469	·463
Correction for form expressed as a percentage of permissible length of standard form at subdivision stations.	Subdivision stations	AT.	+ 5·0	+7·5		+ 5·8
		15%	+ 2·3	+9·9	+ 7·9	+ 5·2
		20%	+ 1·5	+7·5	+4·2	+ 4·7
		30%	+ ·5	+2·0	+ 3·5	+ 4·9
		40%	+ ·2	- 1·2	+ ·5	+ 5·5
		45%	+ ·8	- ·4	+ 1·7	+ 2·5
		50%	+ 2·4	+ 1·6	+ 5·0	+ 2·0
		60%	+ 2·8	+ 4·5	+12·1	+ 2·7
		70%	+ 1·6	+ 2·1	+ 15·7	+ 2·5
		80%	- ·6	- 4·0	+ 5·5	- ·7
		85%	- 1·5	- 6·0	- 7·5	- 2·2
		F.T.	- 1·5	- 6·0		- 2·5
Coefficient of mean waterplane.			·804	·833	·863	·871

ARTICULARS FOR TYPICAL CASES.

	D	E	F	H	I	Q	Y	d
	485	418	448	406·5	335	245	410	650
	500	418	460	406·5	335	245	422·5	675
	59	54	56·08	52	47·5	37·5	55·79	84·5
	45	31	34·08	31	26·25	21	31	48·5
	42·8	30·79	33·87	30·79	26·21	20·67	30·85	48·39
	11·0	9·50	10·50	10·0	6·0	4·75	9·00	10·76
	4·0	4·75	4·00	5·0	3·5	9·25	4·50	6·50
	29·25	25·98	27·67	25·46	19·85	14·88	25·38	34·00
	·704	·719	·741	·723	·672	·719	·681	·648
00	1,006,500	547,100	711,700	520,300	314,100	159.200	544,500	2,021,200
	·796	·789	·815	·803	·755	·838	·749	·732
	·092	·070	·072	·080	·081	·119	·068	·084
9	209·92	175·38	206·85	180·67	137·00	85·32	136·50	292·75
6	81·50	90·54	78·92	86·66	80·17	42·68	132·17	137·5
6	208·58	152·08	174·25	139·17	117·85	117·00	155·85	244·75
30	418,200	222,400	328,900	229,300	113,400	55,000	131;800	812,500
00	202,100	149,400	147,700	136,800	96,400	52,950	222,400	545,800
50	386,200	175,300	235,100	154,200	102,500	71,250	190,100	662,900
	67·4	63·5	63·4		73·5	63	63	68·65
	80	80	80			80	80	86·92
	68·9	64·9	65·4	65	82	68	65	70·26
	·463	·192	·224	·209	·322	·389	·215	·423
	+5·8	+7·6	+4·6		+10·1	-2·0		
	+5·2	+11·5	+4·3		+16·9	-2·5	+16·8	+9·15
	+4·7	+7·8	+3·4		+16·1	-2·6	+9·3	+11·16
	+4·9	+5·1	+2·2		+14·1	-2·4	-1·1	+11·8
	+3·3	-1·2	+1·7		+10·0	-·4	-2·1	+6·0
	+2·5	-·7	+1·8		+9·0	-1·7	+1·3	+4·05
	+2·0	+1·7	+·9	+1·6	+8·0	+1·6	+3·2	+4·8
	+2·7	+3·9	+1·2	+6·0	+7·6	+3·2	+10·4	+6·3
	+2·3	+3·3	+1·0	+6·1	+12·1	+6·0	+10·4	+5·8
	-·7	-·9	-·2	-·7	+4·3	+·7	-4·3	-2·4
	-2·2	-3·0	-1·4	-6·1	-2·0	-2·6	-11·4	-3·3
	-2·3	-7·8	-2·7	-10·3	-2·1	-2·6		
	·871	·830	·873	·840	·834	·831	·808	·808

TABLE 7. OPTIMUM PROPELLER EFFICIENCIES

FROM PUBLISHED RESULTS OF OPEN WATER TESTS

P = S.H.P. AT PROPELLER ; U = DELIVERED HORSEPOWER ; N = R.P.M ; d. = DIA. IN FEET
V_A = SPEED OF ADVANCE IN KNOTS ; $B_{P4} = \frac{N\sqrt{P}}{V_A^{2.5}}$; $B_{U4} = \frac{N\sqrt{U}}{V_A^{2.5}}$; $\delta = \frac{Nd}{V_A}$; Ep = EFFICIENCY %

B_{P4}	TAYLOR AVERAGE				TROOST TYPE A 4.40				TROOST TYPE B 4.40				TROOST TYPE B 4.55			
	B_{U4}	Ep	δ	PITCH RATIO	B_{U4}	Ep	δ	PITCH RATIO	B_{U4}	Ep	δ	PITCH RATIO	B_{U4}	Ep	δ	PITCH RATIO
5					4·40	77·4	93	1·30	4·39	77·0s	93	1·30	4·36	76·3	93	1·32
6	5·13	73·2	100		5·23	76·1	100·s	1·24	5·22	75·7s	101·s	1·21	5·20	75·0	100·s	1·26
7	5·97	72·8	107	1·13	6·05	74·8	107·s	1·18	6·05	74·8	109·s	1·14	6·01	73·7	107	1·21
8	6·80	72·2	113·s	1·08	6·86	73·6	114·s	1·13	6·87	73·8s	117·s	1·07	6·81	72·5	113	1·18
9	7·60	71·4	119·s	1·05	7·66	72·6	121	1·09	7·68	72·9	124·s	1·02	7·60	71·2	118	1·14
10	8·40	70·6	125	1·02	8·46	71·6s	126	1·06	8·49	72·1	131	·98	8·37	70·1	123·s	1·11
11	9·19	69·8	130·s	·99	9·25	70·7	132	1·03	9·28	71·3	137·s	·94	9·15	69·0	128·s	1·08
12	9·97	69·0	136	·97	10·02	69·7	137·s	1·00	10·08	70·5s	143·s	·91	9·90	68·0	133·s	1·06
13	10·73	68·2s	140·s	·95	10·78	68·8	143	·98	10·85	69·7s	149·s	·89	10·65	67·0	138·s	1·03
14	11·50	67·5	145	·94	11·55	68·0	148·s	·95	11·63	69·0	154·s	·87	11·37	66·0	143·s	1·01
15	12·25	66·7	149	·93	12·30	67·3	153	·93	12·39	68·2s	159·s	·85	12·10	65·1	148	·99
16	12·98	65·9s	153	·92	13·03	66·6	158	·91	13·14	67·5s	164·s	·83	12·81	64·2	153	·96
17	13·72	65·2	157	·91	13·80	65·9	163	·89	13·90	66·9s	169	·81	13·53	63·4	158	·94
18	14·45	64·5	160·s	·90	14·55	65·2	168	·87	14·66	66·3s	173·s	·80	14·24	62·6	163	·91
19	15·17	63·8	164·s	·89	15·28	64·6	172·s	·85	15·41	65·8	178	·78	14·95	61·9	168	·89
20	15·9	63·1s	168·s	·87	16·00	64·0	176·s	·84	16·16	65·2s	182	·77	15·66	61·3	173	·86
21	16·6	62·5	172	·86	16·71	63·3s	181	·82	16·90	64·7	186	·76	16·36	60·7	178	·84
22	17·3	61·8s	175·s	·85	17·41	62·7	185	·81	17·63	64·2	190		17·06	60·2	182·s	·82
23	18·0	61·3	179		18·11	62·1	189	·79	18·35	63·6s	193·s	·75	17·75	59·6	187	·80
24	18·7	60·7s	183	·84	18·80	61·5	193	·78	19·07	63·1s	196·s		18·45	59·1	191·s	·78
25	19·4	60·2	187		19·51	61·0	197	·77	19·78	62·6	199·s	·74	19·15	58·6	195	·77
26	20·1	59·7	191	·83	20·2	60·5	200·s	·76	20·5	62·0s	202·5		19·85	58·2	198	·76
27	20·8	59·2	194		20·9	60·0s	204	·75	21·3s	61·5	206	·73	20·5	57·7	201·s	·75

TABLE 8

EFFECT OF PROPELLER REVOLUTIONS ON EFFICIENCY

GIVEN AS PERCENTAGE DIFFERENCE IN S.H.P. FROM 80 R.P.M. BASIS

based on Fig 215 "Speed & Power of Ships" by Taylor.

V_A = speed of advance of screw through surrounding water in knots.
U = useful horsepower or the power delivered by the propeller

V_A	U = 500				U = 1000				U = 1500			
	60 R.P.M.	100 R.P.M.	120 R.P.M.	140 R.P.M.	60 R.P.M.	100 R.P.M.	120 R.P.M.	140 R.P.M.	60 R.P.M.	100 R.P.M.	120 R.P.M.	140 R.P.M.
7	-5·0	+3·2	+9·8	+14·2	-6·5	+5·6	+11·2	+16·3	-6·9	+6·3	+12·0	
8		+4·0	+7·7	+11·4	-5·2	+4·7	+9·5	+14·0	-5·8	+5·4	+10·6	
9		+2·8	+5·6	+8·6	-3·8	+3·8	+7·8	+11·7	-4·7	+4·5	+9·2	+13·8
10		+1·5	+3·5	+5·7	-2·4	+2·9	+6·0	+9·3	-3·5	+3·6	+7·7	+11·6
11								+6·8	-2·3	+2·7	+6·2	+9·3
12										+1·8	+4·5	+7·0
13										+1·0	+2·7	+4·7

V_A	U = 2000				U = 3000				U = 4000			
	60 R.P.M.	100 R.P.M.	120 R.P.M.	140 R.P.M.	60 R.P.M	100 R.P.M.	120 R.P.M	140 R.P.M.	60 R.P.M.	100 R.P.M.	120 R.P.M.	140 R.P.M.
7	-7·2	+6·6			-7·9							
8	-6·1	+6·0			-7·0	+6·4	+12·0					
9	-5·0	+5·3	+10·3		-6·1	+5·9	+11·1		-6·8			
10	-4·0	+4·5	+8·8	+12·9	-5·1	+5·2	+10·0	+14·2	-5·9	+5·4	+10·3	+15·0
11	-3·0	+3·6	+7·2	+10·7	-4·0	+4·4	+8·6	+12·3	-4·9	+4·7	+9·0	+13·3
12		+2·6	+5·6	+8·4	-2·9	+3·6	+7·0	+10·3	-3·9	+3·9	+7·7	+11·6
13		+1·6	+3·9	+6·1	-1·7	+2·7	+5·4	+8·3	-2·8	+3·1	+6·3	+9·8

results into line. In drawing up a rule, it is necessary to test it against known facts, and to see that it is in reasonable agreement with them. The Author's experience, based on an analysis of all the evidence relating to horizontal vibration for one particular case, is that the effect of the entrained water can be taken into account by using the same vertical mass factor for the horizontal as for the vertical case. It would be foolish to generalize upon this one example, however, and still more foolish to put forward untested theories. The conditions are known to be widely different from the vertical case, and such information as is given here may not be generally applicable. In the present state of knowledge it is advisable to allow a good margin on estimates of horizontal vibration frequency; and furthermore, vibration of an amplitude measurable with that at the critical must be expected over a range of plus or minus 5 per cent at least. This is distinct from vertical vibrations, where the tuning is normally very sharp. The greatest service the ship designer can do to assist in reducing the uncertainty is to make common knowledge any relevant data.

PRODUCTION ENGINEERING.

This introduction to the essential factors and principles of production engineering, written by Earle Buckingham, of the Massachusetts Institute of Technology, is reprinted from the June, 1941 issue of Mechanical Engineering. This article is the introductory chapter of a new book on the subject, soon to be published by John Wiley and Sons, Incorporated, and was printed in Mechanical Engineering with their permission.

The problem of production engineering has three major phases: First, the preparation for and the starting of the production of a new product or of a major change in the design of an existing product. Second, the orderly and effective operation of the plant in the continuing production of the product. This involves minor changes in design, generally made to facilitate manufacture, together with changes and improvements in manufacturing processes and the introduction of new manufacturing processes. The problems created by changes in the amount of production, both increases and decreases, are met here. Third, the continuing and supporting activities that gather information and put it into effective form for its direct application to the two preceding phases.

Modern production today involves an almost infinite number of petty details which must be taken care of by the cooperative effort of many persons. The orderly handling of much detail necessitates setting up a definite routine for most of it. It should be remembered, however, that such routine is established to handle the ordinary, or usual conditions; unusual cases will always require special consideration. Or, to put it in other words, routine is a tool or method devised to assist us in the control of a mass of detail; yet any particular detail that can be handled more effectively by other methods should be so handled. To maintain routine, however, anyone so departing from the established routine should be prepared to prove that said departure is justified.

The essence of modern production is the breaking down of the productive effort into simple and elementary tasks, so that less skill, or rather a lesser range of skills, is required of the machine operator. To attain this end, as much as possible of the skill and technique formerly brought to the work by the craftsman must be supplied by the equipment. The further this substitution is carried, the greater are the skill and technique which

must be brought to the problem by the production engineer, the machine maker, the tool maker, and the other craftsmen who design, build, and set up this productive equipment.

The many details of production engineering likewise tend to be divided into more and more specialties, so that when a task of any magnitude is to be done, a large group of specialists must be organized to handle it. When time is the important consideration, as is usually the case in preparation, complete cooperation must exist between them, unless chaos and seemingly endless delays are to result. To my mind, the essentials of effective cooperation are four. First, all persons in responsible charge must have a full realization of the objectives, and the same understanding of them. Second, the cooperating persons must have some understanding of the nature, responsibilities, difficulties, and limitations of the other persons task. Third, each responsible person must be a master of his own specialized task. Fourth, and not least by any means, each responsible person must have a definite understanding of his own responsibility and authority.

It seems axiomatic that responsibility and authority are indivisible. No person or organization can justly be held responsible for conditions over which they have no control. On the other hand, any delegation of responsibility carries with it only that authority necessary to carry out the specific task assigned. The form of an organization and the resulting delegation of responsibility is often determined by the aptitude and experience of the individuals forming the group, yet certain factors of production engineering logically involve certain responsibilities, and should automatically carry with them certain controls, or authority, regardless of the detailed form of organization, if responsibilities are to be effectively met. One of the primary purposes of this text is to point out the fundamental lines of responsibility and authority involved in this maze of production-engineering activities. No claim is made that this analysis is complete, or infallible. Varying combinations of tasks may be assigned to departments and their sections to achieve equivalent results. The attempt will be made, however, to develop a logical analysis of cause and effect.

Let us start the consideration of the subject of production engineering by listing some of the more important factors of the problem, beginning with the initial preparation for production.

Preparation for Production.

1 *Functional Design*—The first essential of production engineering is to have a product to make. The initial design of this product will be called the functional design. In its development, the major objective is to make a product that will perform some function or render a definite service. Some thought may be given here to possible methods of manufacture, yet the greatest emphasis is on its ultimate use in the hands of the consumer.

This functional design may be developed in an existing plant, or it may be brought to it from some outside source. It is the responsibility of the management, assisted of course by information and advice from other parts of the organization wherever they may be found, to decide whether or not to accept this new product for production. Certain definite objectives and requirements should be set up at the start:

(a) Performance requirements of the product.
(b) Cost of manufacture.
(c) Quantity, or rate of production, depending upon the size of the potential market.
(d) Cost of preparation for production.
(e) Elapsed time required to start production.

2 *Production Design*—The production design consists primarily of a critical survey of the details of the functional design. These should be

changed when necessary to facilitate their manufacture with the equipment that is, or may be, available. It includes the setting of tolerances or permissible variations in size of the many elements of the component parts; the selection of suitable materials; the simplification of the design wherever possible; and the use of as many existing standard parts and elements as possible. Indispensable as is the functional design, it is the effectiveness of the production design that largely determines the commercial success or failure of the new product.

It is, or should be, the responsibility of the production-engineering staff, possibly a production-design group, to develop this production design. Whatever is left undone on the production design must be completed, or corrected, after production is under way. It is a process that is never finished as long as the production continues—but the major part of it should be completed before production is started. Such changes to facilitate manufacture should never require the approval of the functional designer; the authority for functional design should have the veto power over any change that would impair the performance of the product. Such objections may be raised over any proposed change, and the resulting arguments, more often than not, are differences of opinion rather than statements of fact. In case of question, it is generally possible to make simple experiments, or in extreme cases, to build a model with the proposed changes incorporated, to prove, or disprove, these matters of opinion.

3 *Estimating*—Preliminary estimates are needed to guide the management to its decision as to whether or not to accept a new product, and also to set up the original objectives and requirements. A more detailed estimate is usually required after the new product has been accepted for production. Such estimates, properly made, serve as a valuable guide to the succeeding work of preparation for production and enable an increasingly larger group to work consistently toward the common objective.

Such estimating is the responsibility of the preparation group of the production-engineering staff. Records and statistics of past performances are an invaluable aid to this work. In general, these estimates give a remarkably accurate forecast of the time and money required for a given project. Too often, however, records of elapsed time, from starting to completion, of the preparation are not kept, and predictions of the actual date when production will be under way, and proceeding smoothly, are much too optimistic.

4 *Operation Layouts and Schedules*—The operation layouts are developed from the estimates, and specify the various machining operations, their sequence, machines, and tools and gages required, and the amount of equipment needed to meet a specified rate of production. This work should also include a schedule, which should take into account the elapsed time needed to design, build, and set up the special equipment. The schedule would establish the order, in point of time, in which each task should be started and finished in order that the specified date for the beginning of production would be met. At times, notes and sketches of some of the unusual features of a tool or fixture should be included to assist the tool designer in his work. In most cases, the locating and holding points on the part should be specified.

This task may also include the making of factory layouts showing the location of the equipment in the plant. The completion of these operation layouts is the responsibility of the preparation group of the production-engineering staff, and this responsibility should continue until the equipment is actually producing the parts in a satisfactory manner.

5 *Tool Design*—The drafting of the jigs, fixtures, tools, gages, and any other special equipment may be done in the tool-design department. This work should be supervised or checked by the process engineer who made

up the specific operation layout. In many cases, these tool designers are specialists on tools for certain specific processes. These designers are responsible for the accuracy and adequacy of their drawings, but the process engineer in charge of a specific component part of the product is responsible for the proper selection of holding points, and the coherence of the series of tools needed to produce that part.

An increasing number of plants is reducing tool-making and tool-designing facilities to the minimum needed for maintenance of current production, so that both the tool designing and the tool making are done by outside companies. In addition, when new machines are required, the machine-tool builder is often required to furnish the machine completely tooled for a specific operation. Here the responsibility of the tool design is delegated to an outside organization. The process engineer in charge of the specific component should then make sure that the correct and necessary information is given to that outside source, since none of his responsibility has been delegated to others.

6 *Requisitions and Schedules for Equipment*—It is the responsibility of the process engineer in charge of a given component to see that orders are issued for the design and the construction of the necessary special equipment, and that schedules showing the sequence and dates when each job should be started and finished are prepared. The clerical work involved may be done in a centralized or general clerical section, but it is his responsibility to see that the section receives the necessary information, and to make sure that the clerical work has been actually done. These requisitions are based on the operation layouts. Questions of priority between the several process engineers should be settled by the chief of the section: Similar questions between the preparation group and the maintenance or production group should be settled by the general management. In this case, we have a problem of interference with routine production by the preparation for a new product, and this is a question of general policy.

7 *Checking of Tools and Tool-Made Samples*—The inspection of new tools, often accomplished by the measurement of work actually produced by the tool, may be done by mechanics and tool inspectors. These men are responsible for the accuracy of their measurements only. The process engineer is responsible for the adequacy of the results. It may be that the same inspectors are used for the checking of tools for new designs and for the routine inspection of replacement tools for the existing production. This, however, does not change the conditions of responsibility.

8 *Initial Production of New Product*—The setting-up, adjustment, and operation of the equipment for a new product may be done by the regular production force, but the process engineer who has planned this work should still be responsible for its performance until it has definitely proved itself in practice. This means that he must give this part of the work sufficient supervision so that all necessary information is in the hands or minds of the proper persons. Definite cooperation is needed here between the planner and the producer. Similar cooperation is usually needed at the very start of the planning.

9 *Checking the Performance of the Initial Assembled Product*—The acid test of the adequacy and completeness of the production design, choice of materials, and all other preparation activities comes when the first of the tool-made parts are assembled and tested for performance. If the product assembles without difficulty, the parts interchange readily, and the assembled product meets all performance specifications satisfactorily, it is conclusive proof that the problem has been solved. On the other hand, if difficulty is met in any of these places, it is equally conclusive proof of incomplete planning, or mistakes, and the preparation group is responsible for these conditions, and must take prompt steps to complete or correct its work.

10 *Investigation and Correction of Initial Troubles*—Experience in the problem of starting production on a new product indicates that some difficulties exist always at the initial stages of production. Some of them are due to ignorance or lack of the necessary special training on the part of the operators; some may be caused by misunderstandings, and lack of full cooperation between the planning group and the production group; some are present because of the incompleteness of the planning; but too often most of them are caused by definite mistakes or ignorance on the part of the planning group. The majority of changes on the part drawings in the initial days of production, most of them made to facilitate manufacture, are evidence of an incorrect, or an incomplete, production design. Unfortunately, many of these escape attention until some of all parts have been made and the first assemblies are on test. But, regardless of the cause of the trouble, the planning group should be responsible for investigating each of them and for correcting the trouble at its source. Close cooperation with the production group will prove invaluable here also.

The foregoing is an attempt at a brief outline of the more important activities involved in the preparation for production. Whether a plant is large or small, these problems will be present. In a small plant, one person or a small group may be responsible for their solution. With a large plant, a considerable organization may be required and the various problems may be divided and subdivided among several sections.

Let us now consider the more important problems of production after what is often a "nightmare" of starting difficulties is well behind us.

Production Operation and Control.

1 *Production Schedules and Follow-Up*—The production requirements are established by the general management, but these are usually in terms of assembled products. These requirements must be broken down, not only into the individual component parts, but also into the individual operations on each part. Schedules must be prepared for this production which will take into consideration all the other work in the plant. All these schedules must be followed up, and hold-ups and conflicts with other schedules reported if predetermined delivery schedules are to be met. Among these component parts will often be found some which are common to several of the products. These are known as stock, or standard parts. They may be made in lots without reference to the specific product orders. It is necessary, in such cases, to determine the economical size of lot to manufacture and the minimum quantity of this stock part to have on hand before starting the manufacture of a new lot. The planning and control of these details are responsibilities of the production engineer. The clerical work involved may be done in some centralized clerical department, but it is the responsibility of the production engineer to see that it is done correctly and on time.

2 *Material Procurement and Schedules*—The material required to make the product is usually ordered by the purchasing agent, but it is the responsibility of the production engineer to see that the purchasing agent has the necessary information to buy the materials required and to keep him informed of the amount needed and when it must be on hand. The amount of any specific material ordered may be based directly upon the production orders in hand, or it may be ordered for stock, as is done with standard parts, when there is any economic advantage in so doing, or if it is a critical material that may be difficult to get at short notice. Such conditions may change with time, and the whole material procurement problem may involve a question of policy which needs the approval of the general management. In this case, the detailed requirements are furnished by the production engineer, the procurement problem is stated by the purchasing agent, and the policy is established by the general management.

3 *Training of Labor*—The training of operators for specific duties is generally the responsibility of the department foremen. The actual hiring may be through a centralized employment department. Such training might well be a part of the process engineer's responsibility. The selection of persons for specific duties should be under the control of the department foreman. The majority of productive operations are relatively simple, and many of them are quite similar. For these the training and assignment of operators should cause little difficulty. In almost every plant, however, there are a few operations that require an unusual combination of skill, temperament, and integrity, and the training of such operators may become an acute problem. Hence the routine training for the ordinary operations may well be left to the foremen, but the unusual case should receive particular attention from the process engineer.

Furthermore, every opportunity should be offered to each operator to increase his skill and value to himself and to the plant. Here, a definite policy, established by the general management, and administered by some part of the production engineering staff, might be worth serious consideration.

4 *Wage Incentives*—To my mind, the most effective wage incentive is one that helps to make the individual realize, to a large extent, that he is in business for himself. Conditions are so varied that the writer doubts if any single system of wage incentive will be the most effective for any given plant. Piecework, bonus, group bonus, and many other types of wage incentives have their place.

Furthermore, wage incentives alone are not enough to secure and maintain the full active support and cooperation of each individual operator. Our concentration on the mechanical advances and refinements tends to push the human factor into the background, despite the fact that this factor is as important as or even more important than the mechanical phase.

The problem of wage incentives is one of policy, to be established by the general management and administered by the production-engineering staff.

5 *Labor Relations*—The subject of labor relations has many aspects, and many books have been written on this subject alone. Among its problems are the following: Seniority, promotion, and security; organized labor (union) relationships or individual agreements; a man's right to a job and a man's rights in his job; and the human relationships between the many individuals in an organization. All, except the last, are largely matters of policy which must be established, to a large extent, by the general management. The last, and to me the most important, is a matter of individual relationships. Improvements in these may be fostered by the management, most effectively by personal example, and self-centered and individualistic persons may be assigned tasks that do not depend on group effort; but for the most part, results really depend upon the behavior of the individuals. We should realize that some valuable potential creative ability is present in every individual of any organization; the great problem is to develop our personal interrelationships so that this potential energy may be transformed into kinetic energy. One of my colleagues remarked, when speaking of assistance given him by mechanics in the shop: "Their field of knowledge may be quite limited, but they illuminate a small spot brightly." In my opinion, the responsibility for improving these human relationships rests with every single member of the organization.

6 *Quality Control*—The quality of any product is tested by its performance in service. Norman F. Harriman, in his excellent treatise on "Standards and Standardization," gives this definition of quality: "Quality, in the sense here used, is that which fits a product for a given use. A product is not simply good, it is good for a certain purpose, and the word quality is meaningless apart from the use in view. Good quality means good for a definite use."

Quality in a product does not develop of itself; it must be definitely and consistently striven for. The effort to achieve it must start with the original design, selection of materials, and choice of manufacturing processes; must continue through all the productive effort, including the assembling and testing; and must in many cases include adequate servicing even after the article is in the hands of the customer. An attitude of "good enough" brings sooner or later a deterioration of quality. Constant effort must be applied to the improvement of quality, always with a view to making the product better for a given use. Changes in design or in processes adopted to reduce costs should always be such as to improve the product.

Quality control during actual production requires a considerable amount of inspection. This inspection may be divided into several phases. For one, we have the preventive measures undertaken to minimize the chances of making mistakes. These include the checking and testing of materials, new tools and machines, and original setups. If the members of the production-design staff are responsible for the performance of their design, this preventive inspection will logically be their responsibility.

For another, we have the process of floor inspection as the parts progress through the several machining operations. The department foreman is responsible for the accuracy of the work produced in his department so that this inspection might logically be his responsibility also. On the other hand, if a general inspection organization exists, the foreman's responsibility should be exercised through the machine adjusters and other assistants, and the routine process inspection should be done by members of the general inspection staff. This does not mean any divided responsibility: The foreman is still responsible for results in his department, while the general inspection staff is responsible for calling to his attention any details overlooked by him or his agents, and for preventing faulty parts from proceeding further.

Following this, we have the finished-parts inspection, to make sure that only correct parts are permitted to flow through to the finished-parts stock room or to the assembly department. This is logically the responsibility of the production design staff.

After assembly, the finished product is often tested for performance, for the making of any special adjustments that may be necessary. Such testing should be logically a part of the responsibilities of the production-design staff, possibly supervised or rechecked by the functional design authority.

One advantage of having the production-design staff responsible for quality control through production is that this responsibility will keep members of this staff in constant contact with the production staff and with many of its problems; and such contact will tend to eliminate the chasm that seems to exist, unfortunately, in too many shops between the drawing room and the shop.

Such quality control should extend to a study of the performance of the product in the hands of the customers. This might be limited to an investigation of complaints from them; but for adequate product development in the future, definite studies of the performance of the product in the field will prove invaluable.

7 *Maintenance of Equipment*—The maintenance of equipment is the responsibility of the staff which uses it. The machine operator or his immediate supervisor is often responsible for reporting the need of repairs. This plan works well in general where there is a single shift of workmen; but when more than one shift is used, the equipment is likely to suffer unless there is a definite organized effort for its maintenance. The responsibility still remains with that part of the production staff which uses it.

8 *Cost Reduction*—Cost reduction efforts include the improvement and rearrangement of the equipment to reduce the productive effort, as well as the introduction of new and improved processes whenever they may become

available. In the original selection and design of processes and equipment we should use the best information available. It is obvious, however, that after production has actually started, we learn much more about its unique problems than we knew before. The slogan for cost reduction effort might well be: "No matter how well we have done a job in the past, it is always possible to do it better and more cheaply." The only question is whether or not we are capable of making that improvement. This is a place where the wholehearted cooperation of the man who does the actual work of production may mean the difference between failure and success.

Cost reduction is the responsibility of the process engineer. He may be attached either to the production staff or to the preparation staff. In fact, the same process engineer may be transferred from one staff to the other, as press of work may dictate.

9 *Correction and Development of Production Design*—The initial production design is nothing more than our first best guess as to the detailed specifications of the component parts of our product that will use the available equipment most effectively, and still retain, and if possible, improve upon, the original functional design as regards its performance. If we can learn by experience, it is clear that after production has started, we should know much more about it; and the more extended our experience with it, the more we should be continually learning. In addition, the initial production design represents in general the opinions and experiences of a small group. As production continues, more and more competent persons become familiar with it. Furthermore, even with the best intentions and reasonable care, mistakes will creep in. All these conditions make it apparent that the development of the production design is a continuing process that is never finished. The production-design staff is responsible for the correctness of this design, and changes should be made without question as their need or value become apparent. If this group is also made responsible for the quality control through production, members of its staff are in constant contact with the progress of events in the shop, and this necessary information will be firsthand knowledge. Otherwise the production group must keep them informed, because a record of these developments is essential, not only to keep the records up to date, but also to have the benefits of this experience which should be incorporated into any new designs.

10 *Cost Control and Budgeting*—Without definite control, the costs of any project tend to mount alarmingly. In order to keep costs within reasonable bounds, it is necessary to be cost- and time-conscious. Probably the best way to develop this sense of time and cost is to budget the estimated amounts available. Each individual and group should then strive to accomplish its specific tasks within the budgeted allowance. If these budgets must be exceeded, it is good practice to require that requests for additional amounts, with the reasons for making them, be made before the original amount is exceeded. In other words, let each explain before the account is overdrawn rather than make excuses afterward.

It should be the responsibility of each group involved to give its own estimate, or agree to the estimate made by others, for the time or money required to carry through any project to a definite stage. These detailed estimates form part of the basis of the estimate that is submitted to the general management. From this information, the general management makes its decision, and sets up the actual budget which is to be followed.

We have now considered the more important problems of routine production. For the solution of these, and also those of the preparation for production, many supporting and continuing activities must be carried on, not all of them an integral part of production engineering, but all having a definite bearing on the problem or its solution. A list of these supporting activities is as follows:

Supporting Activities.

1 *Standardization*—The subject of standardization covers so wide a field that it is difficult to know where to start. It includes the standardization of elementary parts and surfaces, materials, processes, specifications, tools and machines, and methods of test. In one respect, it is the attempt to reduce to routine as many of the elements of engineering as possible. Here, as with routine, these standards are developed to meet the normal conditions, exceptional cases will always need special consideration. Departures from such standards should always be permissible whenever it can be proved that such departures lead to better results than the use of the standards would do.

Standards may be classified under many different headings. Our general engineering standards are formulated under the procedure of the American Standards Association. The majority of our national engineering societies and trade associations have the subject of standardization as one of their major objectives. Almost every large industrial organization finds it necessary to develop specific standards for its own use; while many of them, particularly those which operate several plants, have set up permanent standardization groups in their own organizations. Practically every government department has its standardization group, and some attempt is made to correlate part of this work through such agencies as the Federal Specifications Board. In fact, the development of the production design is, in effect, the standardization of every component part of the product.

In general, any standard must stand or fall on its intrinsic merits. Any standard, to be adequate, must meet the test of utility and economy. Where an adopted standard is applicable, the chances are that in its formulation it has received far more critical study than any individual designer would be justified in giving to a single detail of design, and hence this analyzed solution available should be much better than an inspirational one that may have come on the spur of the moment.

The formulation of standards ought to be a group effort. All interests involved should be represented, the designer, the producer, and the user. The final solution should represent the best judgment of the group as to the best way of reconciling the requirements of functioning with the limitations of manufacturing processes, so as to obtain the maximum utility at the minimum expense. These standards are subject to revision and improvement as experience in their use and production makes evident the need and possibility of further development.

The responsibility for the use of standards wherever possible may be delegated to a standards engineer, but, to be most effective, a definite standards policy should be established by the general management and receive its constant and wholehearted support.

2 *Safety and Accident Prevention*—As time goes on, more and more requirements in regard to safety and sanitation are being specified by law or required by the terms of liability insurance companies. These represent minimum requirements. Some relate to the construction and use of the buildings and building equipment; some to the arrangement of the equipment, aisles, guards over open belts and gears, and some to safety features on individual machines and the attached equipment. The process engineer must do his part in the arrangement of the equipment, or factory layout, and in the design of the jigs and fixtures. The responsibility for checking these conditions may be delegated to a safety engineer on the production staff. His efforts are often checked in turn by periodic inspections of insurance-company representatives and inspectors from the state or local government.

In addition to this, there is the problem of accident prevention. This is largely a matter of individual responsibility, but much may be done by

organized safety campaigns to bring his responsibility home to the individual. One plan in common use tries to invoke the spirit of competition between different departments to keep their accident records clear, or lower than any other department. Similar methods are often used to improve the neatness or cleanliness of departments, aisles, stairways, and washrooms.

3 *Manufacturing Capacity Records*—In order to plan production effectively, it is necessary to have reliable information of the manufacturing facilities available. This requires not only an inventory of the equipment but also of its actual productive capacity, as well as what part is in use on current production. In essence, it is a matter of bookkeeping, and the clerical work required may be done in an order department, cost-accounting department, or in the production schedules department.

The actual productive capacity of any piece of equipment may be quite different from its potential capacity. When a machine is used for short periods of time on successive lots of different parts, the proportion of time required for setups will be greater than when it is used continuously on a fixed setup for the production of a single part. Even in the latter case, its potential capacity is reduced by the time required to change or sharpen cutting tools, cleaning, oiling, and other items of maintenance; so that even with a full-automatic machine, the actual productive capacity may be only from eighty to ninety per cent of the full potential capacity. It is the responsibility of the process engineer to see that such information is collected and properly used.

From this actual productive capacity must be subtracted that part of it already allocated to current production, perhaps leaving a balance which is available for new work. It is the responsibility of the production engineer to see that this information is in the hands of the clerical department which is charged with the responsibility of keeping the records. These records are used by the preparation group in planning new production and by the production group in scheduling current production.

4 *Factory Cost Accounting*—The question of purposes and methods of factory cost accounting is a highly controversial one. The accounting for direct labor and direct material costs is relatively simple, but the equitable distribution of the many indirect expenses is, in general, an unsolved problem. It is safe to say that few, if any, large organizations know accurately their true detail costs. As regards the financial accounting, particularly when a single specialized product is involved, this may be of minor importance; but for the cost-reduction engineer, and the production engineer, this condition at times is serious, because a change is often made to reduce costs, possibly a rearrangement of equipment, which reduces the indirect expense, but because of the method used to distribute this expense, the actual saving is not directly reflected in a reduced-cost record of the part in question. It should be possible, however, to so arrange the cost accounts to serve both the needs of the financial department and the needs of the production department.

Cost-accounting procedure is usually set up by the financial branch of the organization. The information used comes from many sources, including some of the production records. The preparation group has need of considerable statistical information from these accounts to prepare estimates on new products intelligently. The production group should have current and prompt reports from these accounts to know how well it is meeting its budget. The production group is responsible for the accuracy of the information which it sends to the factory cost-accounting department for record.

5 *Time and Motion Study*—Time and motion study is a tool used for many purposes, and many books have been written dealing with its technique and use. It may be used to establish a base for an incentive wage payment on

some new or altered operation; or as a study to check the effectiveness of new or old equipment; or as a study toward the start of an improvement in an old process or arrangement of facilities; or as a test of the skill and increase in skill of one, or of a group of workmen. The results should serve both the production group and the preparation group. The responsibility for this work rests with the process engineer, who may be a specialist in this particular task.

6 *Labor and Equipment Studies*—In too many plants, the accuracy of the work normally produced is a matter of hope rather than of definite information. Many tolerances are established on the assurance of the production department that they can be readily met, which later must be increased materially to suit the actual conditions. One of the responsibilities of the process engineer should be to make a definite study of the limits of accuracy actually attained under the different conditions of operation, and of the degree of skill required of the machine operator as the requirements become more exacting. The preparation group should have a definite table of tolerances which can be maintained on the different types of equipment, and should be given some intimation of the increased care and skill to be exercised by the operator when smaller tolerances are specified. The effort to meet these smaller tolerances will add to the cost. These studies might also include ways and means of training operators to increase their skill. With these studies should be included a tabulation of exact sizes and forms of those elements of the equipment which govern the design and size of fixtures and cutting tools.

7 *Basic Factory Planning*—There are many incidental problems connected with any plant layout, such as provision of steam, gas, water, compressed air, or electric power. If special lines must be installed, and connected to some distant source for each new or changed layout, the expense and time may be considerable. There is always the possibility of making basic, or standard factory layouts such that any of these installations will follow a general basic plan, with provision for additional outlets near where they may be needed in the future.

In general, there are two arrangements which may be used to locate the machinery in a plant: First, the grouping of similar types of machinery together; second, the arrangement of the machines in the sequence in which they are needed to machine a specific component part of the product. The first arrangement is often used when the parts of the product are small, and their transportation from department to department does not create a major traffic problem. The second arrangement is often used on large and continuous production of heavier parts. Here we must often treat our parts- and materials-handling problems as a major task. Here too, some general basic plan as a guide will lead to economy and more consistent results.

All these and many other basic details of factory planning could well be the responsibility of the plant engineers' group of the production department. These basic plans, coupled with definite suggestions for the solution of specific problems of the plant engineer should guide the process engineer in his arrangement of specific shop layouts.

8 *Process Development*—The unique requirements of some part of the product may require the development or the material improvement of some manufacturing process. Again, the success of a new product may be largely dependent upon the development of a new process for certain critical operations. The reverse may also be true—the development of a new process may make possible the development of a new product. In addition, we have always with us the problem of finding better ways to do old jobs. Sometimes the expense of producing some part of our product leads to a search for new and improved methods, involving the development of new processes to meet these needs. Such work may be done intermittently by a specially

organized group attacking a specific problem. In some plants, a definite organized effort is being made continuously to improve existing processes and to develop new ones by a permanent group. This is a part of process engineering, and should be a responsibility of the preparation or experimental group. It should never be a responsibility of the routine production group because routine production and experimental work do not mix, but rather interfere with each other.

The solution of such problems often requires research and experimenting. A laboratory solution may be found, which must next be translated into an experimental factory process, and finally be introduced into the plant as a routine production process. Care must be taken that such new processes are not forced too soon into the production routine, else the production will become too deeply involved in experimental problems, and the production schedule will suffer.

9 *Product Development*—Any product, in order to maintain a high quality, must be constantly improving. Many large organizations have research laboratories, and a large part of their work is directed toward product development; improvement of the present product and the development of new products. This is particularly true of those industries which had their genesis in the laboratory. This does not imply that all of the work in such laboratories is directed for, and dictated by, commercial motives. Yet even if the commercial exploitation of discoveries is only a by-product, nevertheless these by-products, in the long run, must support the laboratory if it is to continue to operate. The work of such research laboratories may be entirely outside the field of production engineering, but eventually the results of some of this work become part of production engineering, passing through the hands of the functional designer and the production designer.

Some plants, in addition to research laboratories, have experimental departments. New and improved products are here devised, often a dozen of them for every new product which is adopted for production. Such experimental work is often a part of the problem of functional design.

Besides this, there is a large amount of analytical work, including the critical analysis of many elements of mechanism, such as gears, cams, linkages, mechanics of materials; information and analyses which are invaluable to both the functional design and the production design, but most essential to the full completion of the production design. Such analytical work could well be a responsibility of the production design.

There are two types of design which, for want of better terms, we will call the inspirational and the analytical. Inspirational design alone will give us many new things to think about, but the results will not always be practical, or of any great commercial value. Analytical design alone will give us valuable and practical results, improving quality, reducing costs, and extending the field of usefulness of existing types of products, but it will seldom, if ever, give us anything startlingly new. The best results require the inspirational features tempered by the cold reasoning of the analytical investigation.

One of the tasks for which it is most difficult to budget the time, and keep within the budget, is the design of a new machine. One reason for this is that the designer too often tries to introduce a large amount of new development with the specific task of combining a group of mechanical elements to give the prescribed service. Such specific design should be rigidly divorced from any attempts at new developments of common elements. The designer should restrain himself, and use tried and proved elements whenever possible. The problem of development is another task and should be solved by itself, and should most often be restricted to the study and development of specific mechanical elements, rather than to complete mechanisms from the start. In other words, standards should be developed for all of the more usual elements of a given type of product, and these standards should be used.

10 *Performance in the Field*—As noted before, the acid test of any product is its performance in the hands of the customer. Despite this, it is surprising how many designers of some products know practically nothing of its actual performance in the use and misuse it receives at the hands of the customer. True, he may be familiar with the conditions and results of tests made in the manufacturing plant before shipping; but many designers of printing presses, for example, know little or nothing firsthand about the problems of make-ready, operating, and cleaning a press as a matter of daily routine. Again, the unguarded remarks of a typist about some feature of a typewriter with which she was struggling might cause the ears of the designer to tingle. We too often assume that in the absence of specific complaints, everything is satisfactory. We might have the young child of a friend visiting us, whose conduct and manners were anything but satisfactory, but only in extreme cases would we complain.

It should be the responsibility of the product-design authority to either establish permanent contact with the users of their product, or to make periodic surveys of its use in the customer's plant. Here is by far the best research laboratory for the collection of performance information that exists. A relatively small amount of money spent for traveling expenses can reveal more pertinent information than many times that amount spent in setting up and operating an experimental department to uncover the same information.

Thus the problem of production engineering begins and ends with the product. Our designs, plant, equipment, organization, and all our activities are only means to this end—that we can place in the hands of our customers a product that will steadily and dependably render him some service he wants or needs.

INDUCTION-HARDENED CYLINDER BORES.

Although some of the most brilliant accomplishments among recent metallurgical engineering developments have been in the field of surface-hardening, the problem of automatically producing an ultra-hard surface on cylinder bores has until now not been satisfactorily solved. The Budd induction heating process, fully described for the first time in this article, is believed to offer the engineer at last the combination of properties he has long sought for many industrial cylinder-wear applications—non-spalling internal surfaces of up to 69 Rockwell C hardness integrally combined with tough, ductile, machinable body-material, and capable of fool-proof production at high rates of speed. Fred P. Peters and Edwin F. Cone are authors of the article, published in Metals and Alloys for June, 1941, from which this abstract is taken.

Metallurgical engineers have been extremely fortunate in the variety and utility of hard materials and surface-hardening processes that have become available to them in recent years. Hard-surfacing metals of the Stellite type, nitrided steels, cemented carbides and borides, flame-hardening, hard-chromium-plating, and induction-hardening of outer surfaces are among the best-known of these recently developed aids to better design and manufacturing practice.

The newest addition to this field is the process for induction-hardening the *internal* surfaces of cylinders, tubes, pipes, etc., developed by Howard E. Somes. Although akin to the foregoing developments in being a method of producing a very hard surface where required for wear resistance, the new method is none-the-less unique in several respects: It is the first

commercially successful application of automatic induction-hardening to the inside surfaces of cylindrical parts—an entirely different proposition from external surface hardening; the hardnesses produced in both steel and cast iron are remarkably high, yet are accompanied by unexpected machinability, physical properties and microstructure; ultra-high integral surface hardness can thus be conferred on certain types of industrial surfaces where it has been most sorely needed—diesel engine cylinder bores, aircraft engine cylinder barrels, wheel hub interiors, hydraulic cylinder barrels, etc.; and the equipment used is actually a precision machine tool capable of turning out hardened parts on a mass-production scale.

This new process, in effect, makes available a new type of engineering material and permits new designs that are basically cheaper to produce. With it there is provided a combination of characteristics not hitherto available in any one metal—extremely high hardness (up to 69 Rockwell C), concomitant wear resistance, exceptional physicals, minimum distortion, and unexplainably adequate machinability. It will be particularly useful to engineers seeking wear resistance combined with high physicals, or for applications where it is desirable to incorporate a wear-resisting surface as an integral part of a load-bearing member. *The process is specially important today because it can turn out fully-hardened parts with the utmost rapidity, and affords a means of saving on defense-necessary alloys.*

The Process and Equipment.

The fundamental purpose of the process is to develop a hard, wear-resisting surface on the inside area of a cylinder without disturbing the toughness, ductility, machinability and dimensions of the main body. A corollary objective has been the production of this condition in "ordinary" steels and irons as well as in alloyed material. Actually, the process is now doing these things in large-scale applications, and an automatically operated control system insures for each job uniformity of area treated, of depth of hardness penetration, and of hardness along the treated surface.

This "selective" hardening is accomplished through the concentration of high-power, high-frequency electromagnetic current in the surface to be hardened. In the process, the high-frequency currents are caused to flow almost entirely in this shallow internal surface zone, and heating is so fast that this zone is raised to hardening temperature almost instantaneously—at least before any significant amount of heat is lost to the main body of the part. The immediate application of a controlled water quench finishes the metallurgical treatment. Close control of results is obtained through accurate automatic control of the power input and frequency of the current, and of the volume, pressure and angle of direction of the water quench. The time taken for the complete heating and quenching cycle is only a few seconds.

The equipment used in the process comprises a source of high-frequency current (usually a motor-driven inductor alternator) and its metering, conducting and transmitting accessories; the hardening machine, with its heating "head," quenching fixture and work-holding device; and the all-important controller system.

For example, one generator in use on this process is hydrogen-cooled and develops a 9600-cycle current, with a capacity of 500 kilowatts at 800 volts. The generator in any case may be mounted in a position remote from the induction-hardening machine—by conducting the current to the machine with a 2-conductor lead-sheathed concentric stranded cable, power losses are no more than 1 or 2 per cent in 500 to 600 feet.

The Hardening Machine.

The cable connects at the hardening apparatus with a specially-designed sliding-core switching transformer, which reduces the voltage to safe working values and also serves to switch the power on and off the heat treating head. The load is applied by moving the transformer primary into inductive relation with the secondary, which is itself connected to the heat treating head by means of a massive conducting arbor similar to the spindle of a machine tool. This arbor consists of 2 concentric copper tubes insulated from each other. The transformer is at the top of the hardening machine, with the arbor and heat head suspended from it.

For each job a suitable work-holding fixture is provided (usually on the lower part of the machine), and in some cases the part being treated is simultaneously rotated and moved axially with respect to the heating head during the treatment-cycle. In operation, it is necessary that the work-piece be chucked so as to be held in the exactly-desired relation with the head, yet not touch it.

The heating head is *the* vital part of the machine. Connected with the transformer secondary, this head comprises a copper tube coiled around a laminated iron core, all on a mandrel. Cooling water is circulated through the tube, since current densities as high as 200,000 amperes per square inch are reached in some cases.

The quenching device operates directly under the heat head, and in controllable relation to it. It is attached to the lower section of the machine, rises toward the head during the treatment cycle, and is retracted downward during loading and unloading.

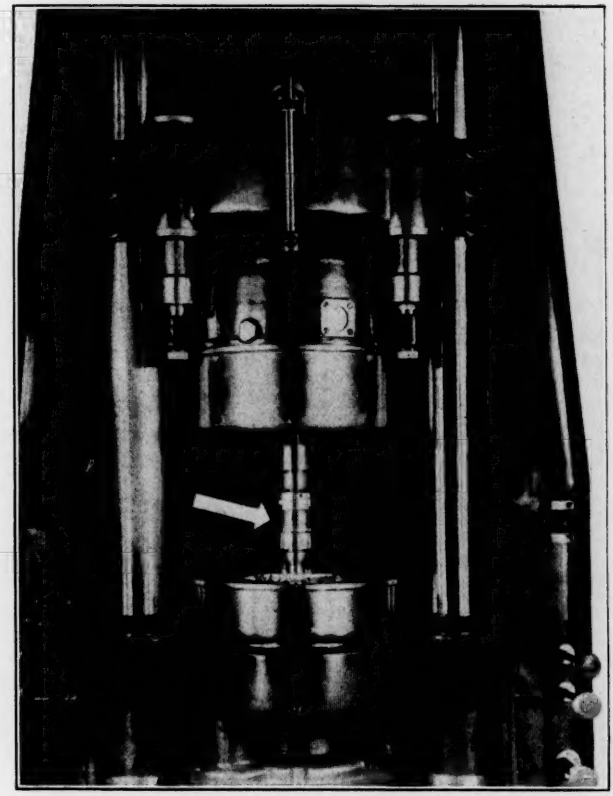
All of these critically important timed sequences are maintained by a sequencing controller made up of a series of precision cam-operated switches, with different sets of cams used for controlling different sizes of cylinders to be treated. Time-consuming change-overs from one bore size to another are thus unnecessary. The controlled movement of the work piece and the heat treating head is accomplished by a motor-driven hydraulic system, and even the coolant passing through the transformer coils is metered.

The machines are compact. Not more than 35 square feet of floor space is required per unit, including "elbow room" around it. At the Budd plant, 2 of these machines are set up and operated in the production line just as any boring mill might be; no stock accumulation is required, and the machine does its job just as fast as the line delivers parts to it.

Thus—in operation—the machine is set for any desired heat treating operation within its range by a convenient adjustment of the controller and its cam-operated switches, and the proper heat head and work-holding fixture; the latter closes and automatically raises the cylindrical work-piece so that it surrounds the heating head, with the latter (and the quench-fixture) in position at the *bottom* of the bore to be treated. The cylinder is then automatically lowered while the high-frequency current is applied, and the quench follows at the correct interval, the treatment thus being "from the bottom up." At a pre-determined moment, the quench is automatically shut off, moved away from the heating head, and the cylinder is removed. The operation—precisely reproducible for an indefinite number of repetitions—may require no more than 3 seconds for the complete cycle.

Applications.

The process has already found important large-scale application for at least 3 types of products, has been regularly used on a small scale for several others, and will certainly be applied on still others that cannot be discussed because of their bearing on national defense.



THE INDUCTION HEATING "HEAD" AND QUENCHING DEVICE, IN OPERATING POSITION, ON A BUDD MACHINE USED FOR INTERNAL SURFARE-HARDENING CAST IRON CYLINDER LINERS. THE LINER (NOT SHOWN) IS HELD IN PLACE BY CHUCKS, ONE OF WHICH MAY BE SEEN JUST BELOW THE HEAD, WHILE THE HEAD IS DRAWN PROGRESSIVELY THROUGH THE BORE.

The original machines were employed for hardening the internal area of forged steel automobile rear-wheel hubs so as to provide an integral roller-bearing race. Since then many new machines and process-cycles have been developed. The most notable of these later applications have been the hardening of cast iron diesel cylinder liners for tractors and the hardening of steel aircraft-engine cylinder barrels. All of these will be described in subsequent paragraphs. In addition, the method is being used for hardening the bores of hydraulic cylinders and of slush-pump cylinder liners for pumping colloidal mud in oil-well drilling, as well as for heat treating oil-well casings to increase joint strengths.

In each of these instances the new method has demonstrated its superiority over the established design or production practice for the product in question —flame-hardening in one case, over-all heat treatment and localized quench in another, nitrided alloy steel in a third, hard-surfaced layers in a fourth, and so on. Yet the new process is far from a cure-all and is not claimed to be the general answer to run-of-the-mill surface-hardening problems; replacement of an older method or design has always been on a basis of quality and serviceability for the particular application involved—usually cylinders and bores that need high hardness backed up by *unusual* physical properties.

In all the applications the convenient machining practice made possible with the new method has been a large factor in its selection. Thus, where design permits, the outside surface or body of the unit can be machined *after* hardening the bore. One new possibility is the practice of composite heat treatment—hardening a bore after the cylinder has been heat treated throughout and machined to finished dimensions. But the most remarkable feature from the machining standpoint is the fact that internal surfaces hardened to 66-69 Rockwell C by this process can themselves be milled or drilled, if necessary, by using carefully worked-out machining techniques.

Aircraft Engine Cylinder Barrels.

The most recent and one of the most promising of the applications of the Budd process, from the general viewpoint, is its use for hardening the bores of aircraft engine cylinder barrels. At the present time, cylinder barrels of forged S.A.E. 4145 steel (1 per cent Cr, 0.20 Mo) are being regularly hardened by this method for a leading engine manufacturer, with eminently satisfactory results. The case produced is more durable, deeper and more uniform concentrically than the nitrided case previously used, and since there is no composition change in the treatment, the hardened surface does not flake off. Production rates are remarkably high.

In fact, according to engineers, the wide application of this process for aircraft engine cylinder barrels could solve one of our most serious bottlenecks in airplane production. Rates of production as high as 500 hardened cylinder barrels a day per machine could be achieved almost immediately, and the use of highly alloyed steels to develop top hardness could be dispensed with; even S.A.E. 1040 steel can be hardened to a sufficiently high value by this process to serve for modern aircraft engine cylinder barrels.

In the application mentioned, the rough-machined forging is first "core-hardened"—heat treated throughout by oil quenching and tempering—to, say, 32 Rockwell C. Then the cylinder is bored and the bore is rough-honed to a few thousandths of an inch under finished size. The outside surface is then turned concentric with the inside and the ends are faced to length.

At this point the bore is induction-hardened in a Budd machine to 62-64 Rockwell C, the depth of hardness being accurately pre-set at about 0.045-0.050 inch. After hardening, the bore is finish-machined, the outer surfaces

are semi-finished and the bore is honed. Distortion in this treatment is so slight that only 0.004 inch to 0.007 inch honing is required—usually just enough to remove tool marks.

Cast Iron Cylinder Liners.

Probably the broadest type of possible application for the new process is its use for hardening the internal surfaces of cast iron cylinder sleeves. Ultra-high hardness combined with practical machinability—all in an inexpensive material with some pretty good inherent "core" properties of its own—are obtained.

At the plant of the Caterpillar Tractor Co. *all* cylinder liners for diesels are now being treated by this process. The liners range in size from 3¾ to 5¾ in I.D. and 10 to 15 inches long; all are treated in the same machine, which is installed directly in the liner production line. The machine hardens the bore surfaces to the equivalent of 60-66 Rockwell C, and these are subsequently tempered to somewhat lower values. The depth of the hardened zone is about 0.070 inch.

Production rates on one machine are regularly between 75 and 105 hardened sleeves per hour, depending on their size. After Budd-hardening and tempering the bores move along the production line, are honed to the final finish before "surfiding," and the outside surfaces are finish-turned for subsequent insertion in the diesel cylinder blocks.

The hardened surfaces of these liners are machinable with special techniques. Caterpillar is actually doing 4 milling operations through the hardened zone—milling 2 valve clearance pockets and 2 rod clearances, and thinking nothing of it, even though the surface hardnesses are such as to make the average machinist exclaim "It can't be machined!" Also, the distortion introduced by the treatment is so slight that on the surface itself only honing (no grinding) is required for finishing.

The finished sleeves are extremely hard, yet not in the least brittle. They are declared by Caterpillar engineers to be superior in wear-resistance and durability, physical properties and uniformity to those previously produced by furnace heating followed by oil-quenching and tempering. In addition, the new process provides automatic control, production-line adaptability, and production rates far in excess of those formerly reached.

Most of these liners are made from a good grade of plain or low-alloy cast iron, cupola iron being entirely satisfactory. A typical iron for this purpose would run 2.80 to 3.10 per cent total C, 0.60 to 0.80 per cent combined C, ½ to 1 per cent Ni, ¼ to ½ per cent Cr, and have a tensile strength of 45,000 to 65,000 pounds per square inch. Close-grained irons should be used, and for thin-walled cylinders the quality of the iron should be chosen with especial care. With a good grade of iron and the correct hardening conditions as governed by the controller, cast iron becomes endowed in the zone treated with the wearing qualities of a fine steel.

An interesting side-feature of this process as applied to the hardening of cast iron cylinders is its simultaneous use as an additional inspection. Where the part being treated is porous or cracked, an abnormality of the heating current occurs within the piece and manifests itself by melting of the surface or wall adjoining the defect; in the case of porosity marked signs of overheating can be observed. Consequently, defective stock can be easily identified—in fact, defects otherwise discernible only by Magna-fluxing or microscopic inspection are readily revealed in this way. Although defects in steel cylinders are also revealed during Budd-hardening, this inspection phase is particularly applicable to cast iron treatment.

Typical Microstructures in Budd-Hardened Cast Iron Cylinder Sleeves (the Iron Is 3.28 T.C., 0.63 C.C., 2.24 Si, 0.68 Ni, 0.58% Cr). Left—the Unhardened Zone, a Characteristic Cast Iron Structure of Graphite Plates and Phosphide Eutectic in a Matrix of Lamellar Pearlite. Right—the "Transition" Region, Showing Partial Grain Refinement. Bottom—the Full-Treated Zone, With Its Uniform, Fine Grain and Unexpected Lamellar Structure. All Micros at 1500 Magnifications.

Some of the Steel Hub Microstructures at Higher Magnifications. Left, the Unhardened Zone at 500X, Showing the Typical Forging Structure of Coarse Grains of Pearlite Bounded by Ferrite. Right, the Fully-Hardened Zone at 2000X, Revealing a Very Fine-Grained Martensite Together With Small Particles of Retained Austenite.

Automobile Rear Hubs.

The first successful large-scale application of the Budd-hardening process was for hardening the inside surface of rear-wheel hubs for a leading automobile manufacturer. The hubs are of S.A.E. 1045 steel forgings, and the bore is hardened to provide an integral race for roller bearings. To date, over 5 million of these Budd-hardened hubs have been placed in service without a failure being reported.

The operation as carried out on a truly mass-production scale at the Budd plant is typical of the results that can be expected when using the process on plain carbon steel parts. The complete cycle requires 20 seconds, and production is normally 180 hubs per hour per machine, although the machines can be timed to operate as fast as 240 parts per hour, if a plant should ever need them that fast.

For this particular application, hardness tolerances are broad (specifications are 56 to 64 Rockwell C) yet the Budd machine turns out hubs consistently with hardnesses between 60 and 62 Rockwell C. The depth of hardness is about 0.10 inch. The wearing characteristics of such induction-hardened bearing races are striking indeed—side-by-side tests have shown that a Budd-hardened race of S.A.E. 1045 steel at 62 Rockwell C excels in performance a carburized S.A.E. 4615 surface.

The machining practice associated with this hardening operation is unusually interesting. The body is irregular in shape and consists of the hub bore to receive the roller bearing, an adjacent flange for mounting the wheel and the drum, and a tapered bored stem for the axle. All boring and turning is done before induction hardening. The hub bore is hardened, and then the bearing bore is ground (actually to remove tool marks) and finished by honing.

Owing to its higher homogeneity, steel surfaces develop higher hardnesses (as measured by the Rockwell superficial tester, for example) in the treatment than do those of cast iron. Steel parts have been readily hardened to 69 Rockwell C and in depths from 0.025 inch to 1/8 inch, and certain special cast steels have even been treated to hardnesses up to 78 Rockwell C.

Metallurgical Considerations.

From all of the foregoing, it is evident that the process must be metallurgically "different" from more familiar hardening methods, since so many of the product's characteristics that are microstructural in origin are unique. In addition to the unusually high hardness and physical properties obtainable with a given composition, there must be cited the fact that Budd-hardened surfaces are machinable beyond the limits to which other, equally hard, materials are subject. Another interesting feature is an evidently different reflectivity of tooled surfaces that have been treated by the process; it is this that makes possible clear and prominent delineation of the depth of hardness in a cross-section without metallographic polishing and etching.

These points undoubtedly have their origin in the microstructures resulting from the new treatment. The extremely rapid quenching produces in both cast iron and steel a much finer-grained martensite than is ordinarily obtained by other methods. Distortion is held to a minimum because only a portion of the piece is heated to critical temperatures. Absence of scaling, spalling, grinding checks and cracks, and of relief of hardening stresses during subsequent machining or grinding, and, in many cases, the obviating of pre-annealing or pre-normalizing treatments, all stem from the unusual speed of heating and uniformity of structure.

In all cases the grain is ultra-refined, ordinarily there remaining no vestige of the original grain boundaries. In cast irons the uncombined carbon seems to be recombined at a greatly accelerated rate with attendant increase in hardenability and strength of the area treated. One of the most startling features is the lamellar structures that are found throughout the hardened portions of originally pearlitic irons. The lamellae are scarcely visible at ordinary (100-500 X) magnifications, but at 1500 X they closely resemble pearlite in iron at about 500 X.

We are indebted to L. R. Jackson of Battelle Memorial Institute, who has had considerable opportunity to examine the Budd-hardened structures metallographically, for the following information on microstructures. His comments throw considerable light on the reasons for some of the unique and superlative characteristics of iron and steel treated by this process.

In spite of the high hardness (60-66 Rockwell C) of the cast iron cylinder liners there is a considerable amount of retained austenite in the hardened zone. The austenite is evident to X-rays and to magnetic analysis, but does not show up under the microscope in a recognizable form; it must therefore be ultrafine-grained and dispersed throughout the structure. Austenite contents of 30 per cent and above have been observed. It is possible that the presence of the austenite is a factor in the existence of machinability at such high hardness. The combination of retained austenite and high hardness may also contribute to the wear resistance of the surfaces.

The hardened zone in 0.40 per cent carbon steels (S.A.E. 1040, 3140, 4140, and 3340, etc.) that have been examined also shows the presence of more austenite than would be expected from a more conventional heat and quench. Amounts around 10 per cent have been observed.

The induction heat treating of steel or cast iron involves heating times from a fraction of a second to a few seconds. From the viewpoint of the old rule of thumb "soak an hour per inch of cross section" the results obtained with this internal induction heat treatment seem incredible. However, Jackson points out, we can safely assume that the temperatures attained in the Budd treatment are 300 or 400 degrees F. higher than the usual austenizing treatments, and therefore need make no special assumption about the diffusion coefficient for carbon in order to explain the results obtained.

Apparently the induction heating takes the affected zone to a temperature so high that diffusion is rapid, yet the time at temperature is so short that the excessive grain-coarsening that would ordinarily be expected to accompany such high temperature heating does not occur.

Conclusion.

Whatever the reasons, however, observation of the Budd process for induction hardening of cylinder interiors and of the service results it provides, makes altogether clear its position as an established commercial success with considerable still unexploited potentialities. The machine and process, by permitting the fabrication of machine parts of maximum wear resistance without sacrifice of physical properties, present the metallurgical design engineer with a new type of material for his cylinder bore applications and will be a step toward new designs and new products.

The sizes and lengths discussed in the article are in no sense restrictive, either, for machines already built handle lengths from a fraction of an inch up to several feet—and existing designs contemplate unlimited lengths! The new *internal-heating* process constitutes an ideal complement to the well-known systems for *external* surface-hardening by induction, extending the industrial applications of induction-heating to fields where it has been most sorely needed.

HOW TO DETECT AND CURE WIRE ROPE TROUBLES.

In this article, which should be of particular interest to operating personnel, A. J. Morgan, chief engineer of the Wire-Rope Division, John A. Roebling's Sons Company, shows what causes wire-rope difficulties and how to remedy them for long rope life. The article here reprinted was first published in two parts in the February and March, 1941 issues of Power. The two parts are herein combined and accordingly certain figure numbers are duplicated.

Rope Inspections.

Many wire-rope troubles would never occur if the owner of the machine understood the fundamentals of how to make an examination of his equipment, if he understood what the engineer does when he inspects a machine. There is nothing complicated about it, and it is an excellent idea, both for safety and in saving money, to make regular, thorough examinations.

The first thing an engineer remembers is that there are seven principal ways in which a wire (the fundamental unit of a wire rope) can be parted:

1. It can be cut through.
2. It can be pulled apart under tension.
3. It can be worn through by abrasion.
4. It can be eaten through by rust.
5. It can be broken in two by fatigue.
6. It can be twisted apart.
7. It can be mashed.

Consequently, when the rope engineer peers studiously at the individual broken wires, he is trying to establish which combination of the above factors has been responsible. This he can discover from the ends of the broken wires themselves, because each of the seven kinds of break will produce a different appearance in the cross-sectional view of the broken wire.

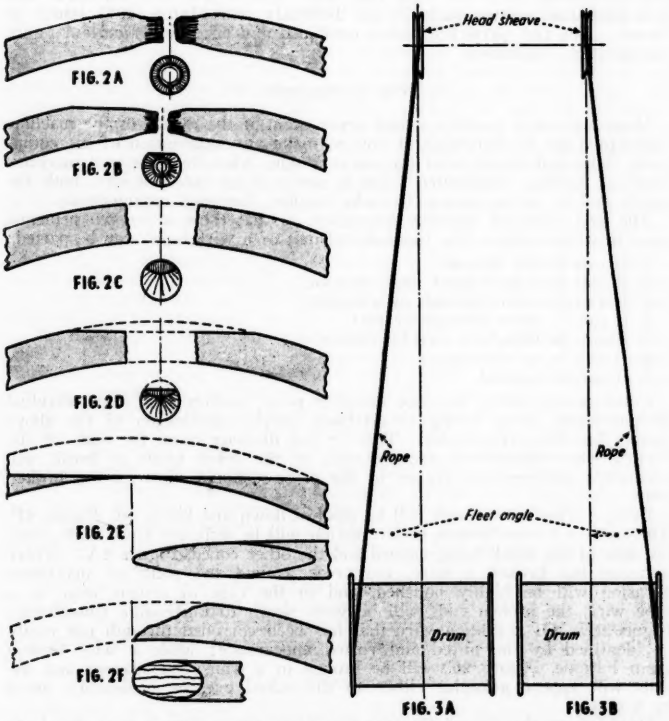
First, a wire cut through will be pinched down and bitten off, Figure 2F. The ends of a wire broken under tension will be different from each other, one side of the break being cupped and the other coned, Figure 2A. Where abrasion has broken a wire, the areas around the point of maximum abrasion will be highly polished, and in the case of crown wear in a rope wire, the broken ends will be worn down to knife-edge smoothness, Figure 2E. Of course, a wire that has been corroded through can easily be identified by the pitted and rusted appearance; while a wire broken from Fatigue, Figure 2C, will be broken in a transverse manner and the ends will appear granular either to the naked eye, or preferably, aided by a glass.

Once having determined the factor at fault, the engineer's next step is to locate the cause of this factor. He makes a thorough study of your machine. It's nothing intricate. He merely checks to see whether a few fundamental working conditions are maintained, and if not, to what extent their absence has been responsible for the poor rope service.

Obviously, even with the right rope on the machine, the rope might break under tension if too heavy a lift is attempted. Thus, no further discussion is necessary on the subject of tension breaks, except to say that if other factors have weakened enough wires and strands, the rest may fail easily under strain of a normal load.

Drum Conditions.

The engineer will examine the drum. Assuming that the right rope is on the machine, the drum diameter should be commensurate with the



FIGURES 2A TO 2F—TYPICAL WIRE FRACTURES: A—CUP AND CONE TYPE, NO ABRASION. B—CUP AND CONE TYPE WITH ABRASION. C—FATIGUE TYPE, NO ABRASION. D—FATIGUE TYPE WITH ABRASION. E—WORN COMPLETELY THROUGH. F—CUT OR GOUGED WIRE. FIGURES 3A AND 3B—FLEET ANGLE WHEN ROPE IS IN ITS TWO POSITIONS OF GREATEST TRAVEL.

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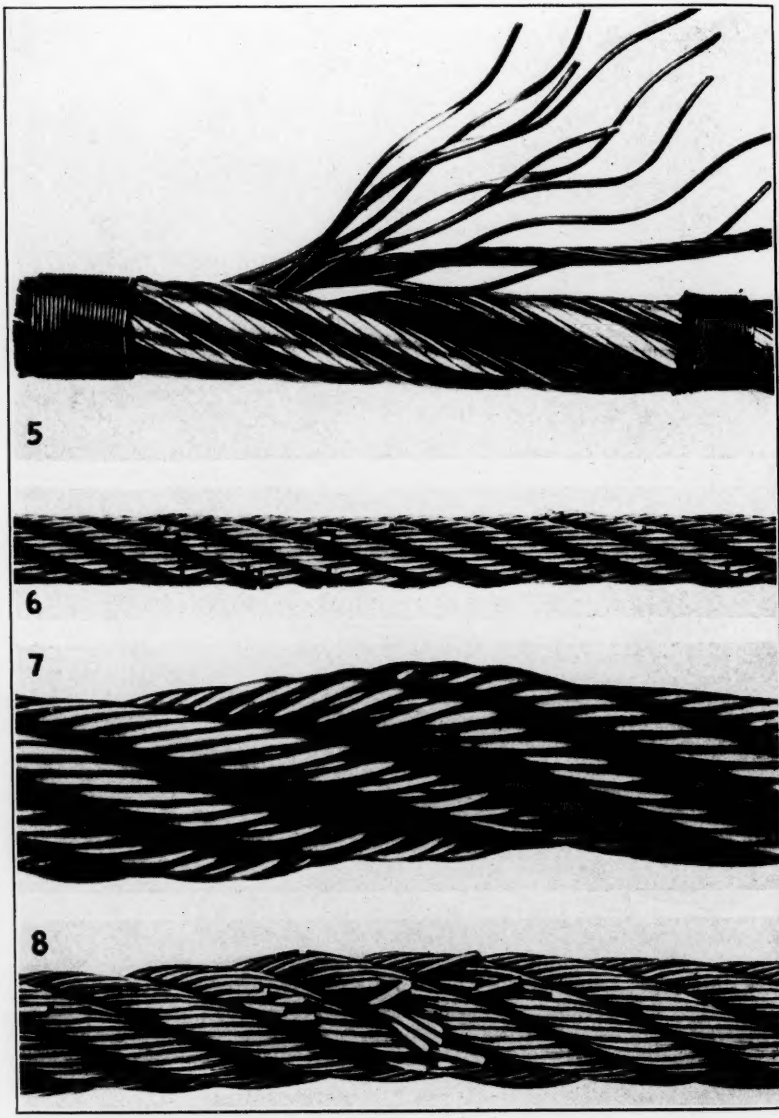


FIGURE 5—ALTHOUGH BADLY WORN ON THE OUTSIDE, THIS ROPE SHOWS NO SERIOUS EFFECTS FROM SERVICE ON THE INSIDE BECAUSE IT WAS WELL LUBRICATED. FIGURE 6—BROKEN WIRES, NOT WEAR, CAUSED THIS ROPE TO BE CONDEMNED. FIGURE 7—ONCE A ROPE HAS BEEN KINKED THE DISTORTION CANNOT BE REMOVED COMPLETELY AND EARLY FAILURE IS IN PROSPECT. FIGURE 8—BROKEN WIRES IN THIS ROPE RESULTED FROM A KINK DURING INSTALLATION.

rope's construction. But, there is more to a drum examination than that. The surface must be checked. If it is grooved, are the grooves of the proper radius to accommodate the rope? Drum grooves are governed by the same rules as those for sheaves: this question will be discussed later.

The next question to settle: is the drum face uniform, or is there a high or low sp.. in its surface? If there is a high spot, the rope will tend to pile up away from the spot, towards the flanges. This will cause undue scrubbing of one wrap against the adjacent wrap as the drum winds. Conversely, if there is a low section, the rope will tend to pile up towards the bottom of that section, away from the flanges. There will then be undue scrubbing again of adjacent wraps as the rope winds on the drum.

Fleet Angle.

Next, we come logically to an examination of the "fleet angle." This term should not, but does, frighten some rope users. The fleet angle is simply that angle between the center line of the "lead" sheave and the rope, Figures 3A and 3B. The lead sheave is the first sheave over which the rope passes as it winds from the drum.

The fleet angle must be kept within certain limits. If it exceeds 1.5 degrees, the rope may wind unevenly, crowding and scrubbing in spots, and winding loosely in others. A large fleet angle will also cause the rope to scrub against the flanges of the lead sheave, adding another source of abrasive wear. Again, if the angle is less than 0.5 degree, the rope may wind unevenly if more than one layer of rope is used.

Of course, these conditions are ideal. Occasionally, in the interest of economy or practical machine design, they must be violated to some extent.

It is important to the life of a rope that it be wound evenly on the drum. This objective is usually reached by having the proper fleet angle. On the other hand, normal care must be taken that the rope originally is wound tightly and in close wraps. Care must be taken also that there is always sufficient tension on the rope, when a load is released. If this is not done, the natural elasticity of the rope will snap back to the drum and cause a misarrangement of the wraps, thereby producing conditions of pile-up and crushing.

Even under normally good drum-winding conditions, there will frequently develop a condition of sectional wear in a rope which winds in multiple layers. Such wear is not, as is sometimes claimed by the inexperienced, caused by a series of "soft spots" in the rope. It is probably occurring at the "cross-over" points or the layer-starting point.

Cross-Over Points.

The cross-over point is that point at which the rope of the on-coming layer crosses over the rope on the layer below. This occurs when the rope is forced, by the wrap next to it, out of the groove formed by two adjacent wraps on the lower layer in which it has been winding. It must then climb over the crest of one of the bottom wraps before becoming engaged in the next groove. The top rope crosses two wraps of the underneath layers in this manner for every revolution of the drum. It is at the spots where this action takes place that constant abrasive action and consequent concentrated wear occur on the rope.

Another somewhat similar instance of concentrated abrasion found in normal multiple layer winding occurs at the point where the on-coming layer is filled and the rope is pushed up to start a new layer. After the first wrap of the new layer is wound, the on-coming rope will strike the point at which the layer was started and be shunted over acutely. At this point abrasion and scrubbing on the surface wires of the rope will be high.

The remedy for these conditions is to unwind the rope occasionally and cut off a piece at the drum end equal in length to 1.25 or 1.75 times the diameter of the drum. In this way, the cross-over points and layer-starting points are changed and the wear is distributed among several different sets of points along the rope instead of concentrating only in one. By all means do not remove a multiple of the drum diameter, because, if such a length is cut, old cross-over and layer-starting points will fall adjacent to each other again and wear will continue at these places.

The engineer, in looking for reasons why rope is short-lived, knows that all of them are not due to the drum or the way the rope winds onto it. He will, therefore, examine the condition of the sheaves.

If he finds that the sheaves are of the proper diameter for the size of rope running on them, he inspects the radius of the rope-carrying grooves. These grooves must be cut to offer maximum support for the rope, without restricting normal rotation. Thus, if a sheave-groove radius to too large, Figure 1, the rope is not properly supported in the groove and, under tension, has a tendency to flatten out. On the other hand, if a sheave-groove radius is too small, Figures 3 and 4, the rope is pinched and an undue amount of rope abrasion against the sheave flanges takes place.

Groove Tolerance.

The groove should support the rope for nearly one-half of its circumference, Figure 2; thus, the groove radius must be slightly greater than that of the rope. The table on page 655 gives the proper tolerance by which sheave-groove diameter should exceed normal rope diameter for proper operation. Wire-rope manufacturers supply gages with which these grooves may be checked.

Results of improper sheave grooving will be, in addition to the distortion mentioned, excessive abrasive wear showing on the sections of rope that pass most frequently over the offending sheave. Thus, if you find that one section of a rope is more worn than others, you can locate the source of trouble, if it is a sheave, by noticing which one this section passes over most frequently in the course of normal service.

Eccentric rotation of a sheave can also be a source of rope trouble. This can be caused either by the sheave itself being out of round in its circumference, or by its being worn eccentric, which will set up a whipping action in the rope. This action will have a severe fatiguing influence on the rope, concentrated in certain spots along its length. These spots, of course, vary in location according to the individual installation, but can easily be recognized by the fact that typical fatigue breaks in the wires will appear and increase rapidly in these sections.

Whipping Action.

It is also possible for such whipping action to cause the rope to slap against some immobile part of the machinery. In such a case there will be evidence in the form of easily recognizable scrubbing and gouging of the wires.

It should be noted, however, that in some instances, a certain amount of vibration is inherent either in the machine itself or in the operation of the machine. Such vibration may cause whipping of the rope, and in such cases whipping is likely to be difficult to eliminate. This is especially true of some excavating machines, and also of some drag scrapers where there is a heavy horizontal pull. Consequently, under these conditions, the maximum protection to the rope should be insured, if possible, by installing "whipping blocks," wooden blocks placed over any hard surface against which the rope might slap.

In installations where there are guide rollers to lead the rope into a sheave, another potential source of trouble exists. If these rollers become deeply grooved by the constant passage of the rope over them the edges of these grooves will cut the rope as it moves back and forth transversely across them. The surface of such guide rolls must be inspected and kept reasonably smooth. Care must also be taken that these rolls are free to rotate, for if they become "frozen," abrasion will increase.

Internal Corrosion.

One of the most difficult sources of rope deterioration to trace down is internal corrosion. A rope in service cannot be opened up for internal inspection throughout its length and there are only two methods of establishing whether or not a rope is corroding inside. The first is inspection at the time cut-backs are made at the drum, or preferably at the bucket end. In this way you can get a fair idea of the condition throughout the rope length.

Where cut-backs are not made frequently, close inspection of the "valleys" between strands must be relied upon. If the rope appears dry and there is evidence of any pitting or rust in these valleys, internal corrosion is to be suspected. If there are wire breaks occurring in the valleys rather than on the crowns of the strands, an inspection of the broken ends will reveal whether corrosion, fatigue, overstressing or cutting is responsible.

Cutting will be the result of either of two conditions. Either there is a sheave with an under-sized groove, which is pinching the rope so tightly that the wires are cutting into each other, or the center of the rope has become so dried out that it no longer is offering sufficient support to the strands, which are cutting into each other.

Groove-Diameter Tolerance.

Normal Rope Diameter, In.	*Proper Groove Clearance, In.*
1/4 to 5/16	1/64
3/8 to 3/4	1/32
13/16 to 1-1/8	3/64
1-3/16 to 1-1/2	1/16
1-9/16 to 2-1/4	3/32
2-5/16 and larger	1/8

Again, a condition of overstressing might have stretched the rope unduly, thereby reducing the diameter to such a degree that the strands have crowded together and allowed the individual wires to cut each other. A hemp center will collapse when it is not lubricated enough.

Where ropes are dragged through sand or any similar granular substance, it is not advisable to coat them with heavy lubricant. This causes the rope to pick up particles which will be ground into the small spaces between the wires and strands and will cause internal abrasive wear which can seriously weaken the rope. On such installations, ropes should be lubricated frequently with a light, penetrating preparation. The outside will probably not be seriously affected by corrosion because of the fact that passage of the rope through the sand will keep the steel bright and rusting will not gain any headway.

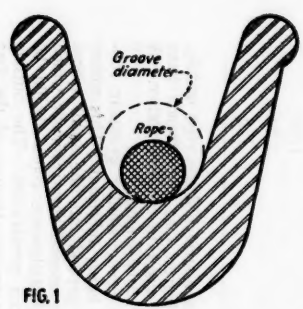


FIG. 1

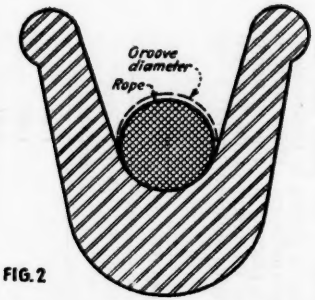


FIG. 2

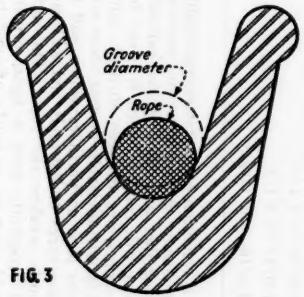


FIG. 3

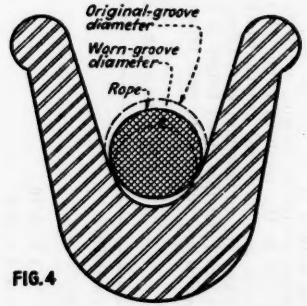


FIG. 4

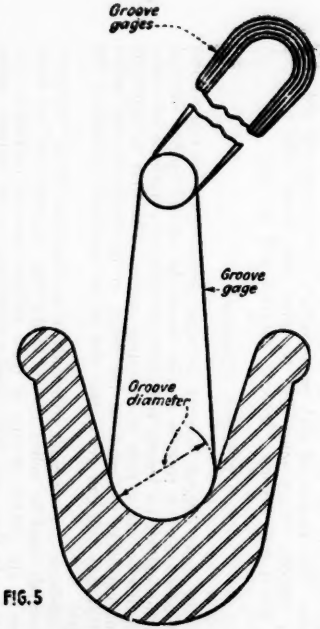


FIG. 5

FIGURE 1—IF THE SHEAVE GROOVE IS TOO LARGE THE ROPE IS NOT PROPERLY SUPPORTED AND IT TENDS TO FLATTEN UNDER TENSION. FIGURE 2—THE GROOVE SHOULD SUPPORT THE ROPE FOR NEARLY ONE-HALF OF ITS CIRCUMFERENCE. FIGURE 3—A CORRECTLY DESIGNED SHEAVE GROOVE WORN TO DECREASED DIAMETER OF THE ROPE. FIGURE 4—NEW ROPE OF NORMAL DIAMETER IN A SHEAVE GROOVE WORN BY PREVIOUS ROPE OF DECREASED DIAMETER. FIGURE 5—MEASURING THE GROOVE IN A SHEAVE WITH A GROOVE GAGE.

There are any number of unique potential causes of poor rope service which must be ascertained at the individual installation. For instance, if the rope comes into constant contact with any part of the machinery, undue wear will result. Or again, sudden shocks to a rope under tension will cause stresses which may produce tension breaks. In short, all manner of mechanical abuse must be taken into consideration when the causes of poor rope service are being investigated.

Short Rope Life.

Having thus examined the rope and installation for all possible sources of trouble, the engineer is really just beginning his job. It is at this point that he must call upon his training and experience. Frequently, a number of conditions will be found on a machine that are not ideal for longest rope life. Yet some of these may have been present during the life of several ropes without having seriously affected their service. Then another factor, possibly equally as minor as those already existing, may enter the picture and tend to aggravate the situation, thus resulting in serious decrease in rope life.

For instance, a sheave may have been a trifle eccentric for some time without being noticed. Then, for some reason, one rope is not lubricated properly and corrosion gets a start. The rope will then be much more susceptible to the fatiguing effect of the whipping set up by the eccentric sheave. Broken wires appear much more rapidly than seems reasonable and the rope gets the blame. The engineer must be able to recognize the way in which the two factors work hand in hand, the one aggravating the other. He must be able to determine whether or not the expense of a new sheave would be warranted if the rope were properly lubricated.

It is impossible to make absolute general statements to cover all possibilities of rope service. Installations and their individual conditions vary as widely almost as do the personalities of the people who own them. The engineer's search for trouble on a rope installation may be summarized: to establish all possible causes of rope deterioration found on the installation or in other sources; to weigh the evidence thus collected in the light of its own value and of his experience; and finally, to determine whether or not the factors considered are sufficient to cause the rope deterioration. If, as we have assumed in these articles, the rope used was the proper one, in the vast majority of cases the cause of under-average rope service will be found in these factors or a combination of them.

CARBON STEELS FOR HIGH-TEMPERATURE SERVICE.

This paper by H. J. Tapsell and A. E. Johnson gives a brief account of the influence of stress, temperature, and time on the behavior of carbon steels of about 0.15 to 0.50 per cent carbon content, and provides data as a basis for design purposes. The data given are derived from investigations carried out at the National Physical Laboratory, largely on behalf of the British Electrical and Allied Industries Research Association. The chief purpose of the paper is to assist the reader in appreciating the factors involved in estimating the useful strength of steels at higher temperatures extending to about 1000 degrees Fahrenheit. The paper is reprinted from the January, 1941, Journal and Proceedings of the Institution of Mechanical Engineers.

FIGURE 2—THE GROOVE SHOULD BE CORRECTLY DESIGNED IT TENDS TO FLATTEN UNDER TENSION. FIGURE 3—A CORRECTLY DESIGNED ROPE FOR NEARLY ONE-HALF OF ITS CIRCUMFERENCE. FIGURE 4—NEW ROPE OF SHEAVE GROOVE WORN TO DECREASED DIAMETER OF THE ROPE. FIGURE 5—WORN SHEAVE GROOVE. NORMAL DIAMETER IN A SHEAVE GROOVE WORN BY PREVIOUS ROPE OF DECREASED DIAMETER. SHEAVE GROOVE WORN TO A CHORD. GAGE.

I. Tensile Properties.

(*a*) *Properties at Room Temperature.*—Before dealing with the effect of temperature on the tensile properties of carbon steels it is considered necessary to give a brief account of some of the variations in properties at room temperature, due to differences in carbon content, and in heat treatment. Generally, one may expect an increase in ultimate stress with increase in carbon content for steels similarly manufactured and in the same final condition, *e.g.*, cast, hot-rolled, annealed, although increases of at least 0.05 per cent carbon may be necessary before an increase in tensile strength is evident. Yield points and limits of proportionality, however, vary considerably for steels of the same carbon content, even when manufactured by the same process, and these properties are not solely a function of the carbon content. This is illustrated by the values in Table 1, from which it will be seen that the yield points and, in particular, the limits of proportionality differ considerably for steels of similar carbon content. All the steels referred to had been given the usual normalizing treatment prior to the tests. It may be mentioned that there is nothing which would be considered unusual in the chemical composition of the steels.

In addition to the variations in tensile properties amongst steels of similar carbon content, certain marked changes in the structures of steels produced by heat treatment may have much less effect on the tensile strength than on the yield points and limits of proportionality. Table 2 shows the differences in tensile properties of two wrought steels, each in three different conditions referred to as normalized, spheroidized, and coarsened, respectively. Heating for 18 hours at 650 degrees C. (1200 degrees F.) after normalizing changes the lamellar structure of the carbide to a partially spheroidized or globular form, the carbide tending to become scattered throughout the ferrite (iron) grains. The crystal grains are rendered larger (coarsened) by heating the normalized steel for 6 hours at 1100 degrees C. (2012 degrees F.) and cooling in the furnace. From Table 2 it will be seen that the ultimate stresses of the steels are not very much reduced by spheroidizing or coarsening. In the case of the 0.13 per cent carbon steel, improved ductility results from the spheroidizing and the coarsening treatments, but the yield points and limits of proportionality of both steels are adversely affected, especially by the coarsening treatment. It should be stated that, although the additional heat treatments were given to the steels particularly for the purpose of examining the effect produced on their properties at high temperatures, both partially spheroidized and coarsened structures may be encountered in steels which are only intended for use at room or moderately elevated temperatures. Forgings and castings of appreciable size are invariably air-cooled and tempered.

(*b*) *Properties at Elevated Temperatures.*—It is a characteristic feature of carbon steels that their tensile strength, as determined by loading tests to fracture at normal testing speeds, shows an increase in the "blue-heat" range, about 200-350 degrees C. (400-650 degrees F.), which is often very marked. The general behavior of carbon steels up to 1000 degrees F. is illustrated by Figure 1, which shows how the tensile strength at various temperatures is related to that at room temperature. Between the upper and lower curves lie all values likely to be obtained for carbon steels of 0.10-0.60 per cent carbon content. The upper curve relates more particularly to steels of low carbon content and the lower curve to steels of higher carbon content; but the speed of testing is an important factor, higher values being obtained above the blue-heat range the more rapidly the load is applied.* Bearing that in mind, the dotted curve can be accepted as sufficiently representative of the average tensile properties of a carbon steel under normal testing conditions.

* Between room temperature and the blue-heat range extremely slow loading tends to give higher results than rapid loading owing to additional time being available for the ageing process which is responsible for the increase in ultimate stress.

TABLE 1. VARIATION OF TENSILE PROPERTIES AT ROOM TEMPERATURE OF VARIOUS WROUGHT CARBON STEELS AFTER NORMALIZING

Steels in form of 1-inch rolled plate

Carbon content, per cent	Process of manufacture	Heat treatment	Limit of proportionality, tons per sq. in.	Yield point, tons per sq. in.	Ultimate stress, tons per sq. in.	Elongation, per cent	Reduction of area, per cent
0·15	Acid open-hearth	15 minutes at 950 deg. C.; air-cooled	19·8	19·8	30·9	43·5	68
0·16	,, ,,	Do.	11·1	15·8	29·0	43	67
0·19	,, ,,	Do.	8·9	14·4	27·8	42	63
0·13	Basic electric	Do.	16·5	17·1	27·2	33·5	58
0·20	,, ,,	Do.	20·2	22·0	31·9	43	67
0·40	Acid open-hearth	15 minutes at 900 deg. C.; air-cooled	18·0	21·9	40·0	32	49
0·40	,, ,,	Do.	20·7	22·2	39·0	32·5	53
0·42	,, ,,	Do.	10·0	—	38·3	28	39
0·40	Basic electric	Do.	23·7	23·8	37·2	33·5	54
0·44	,, ,,	Do.	22·8	23·4	40·3	31	47

TABLE 2. EFFECT OF STRUCTURAL CONDITION ON THE TENSILE PROPERTIES AT ROOM TEMPERATURE OF TWO CARBON STEELS

Steel	Heat treatment	Tensile properties				
		Limit of proportionality, tons per sq. in.	Yield point, tons per sq. in.	Ultimate stress, tons per sq. in.	Elongation, per cent	Reduction of area, per cent
0·13 per cent carbon; 1-inch rolled plate	15 minutes at 950 deg. C.; air-cooled (normalized)	16·5	17·1	27·2	33·5	58
	15 minutes at 950 deg. C.; air-cooled; 18 hours at 650 deg. C.; furnace-cooled (spheroidized)	15·1	15·3	25·4	51	69
	15 minutes at 950 deg. C.; air-cooled; 6 hours at 1,100 deg. C.; furnace-cooled (coarsened)	12·7	13·2	25·8	47	63
0·40 per cent carbon; 1-inch rolled plate	15 minutes at 900 deg. C.; air-cooled (normalized)	23·7	23·8	37·2	33·5	54
	15 minutes at 900 deg. C.; air-cooled; 18 hours at 650 deg. C.; furnace-cooled (spheroidized)	20·7	21·2	35·4	38	56
	15 minutes at 900 deg. C.; air-cooled; 6 hours at 1,100 deg. C.; furnace-cooled (coarsened)	11·9	16·5	35·3	33	41

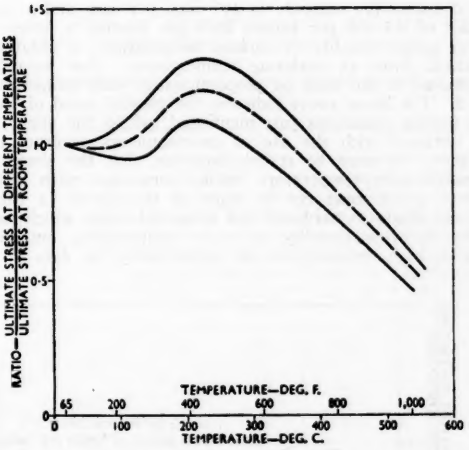


FIGURE 1. CHANGE IN ULTIMATE STRESS OF CARBON STEELS (UP TO 0.6 PER CENT CARBON) WITH INCREASING TEMPERATURE.

The curves refer to cast, wrought, annealed, and normalized steels. Most published results lie between the full curves. The dotted curve is representative of average values.

Increasing importance is attached to the rate of loading (or straining) as the test temperature is increased; a very slow rate of straining results in failure at a much lower ultimate stress than is obtained for rapid straining. Such a result is due to creep or flow of the metal which resists the building up of stress, the more effectively if the rate of straining is slow. The practical significance of creep lies in the continuous deformation produced under steady stress, and this aspect of creep is dealt with later in the paper. Meanwhile it may be stated that the ultimate stress is not a satisfactory criterion of the strength of a steel at high temperatures, for the purposes of design.

The measured value of the limit of proportionality is dependent at all temperatures on the sensitivity of the extensometer employed for its determination. An instrument capable of detecting a strain of 10^{-5} inch per inch will disclose the point at which the strain deviates by more than 10^{-5} inch per inch from proportionality with stress, and naturally registers a lower limit of proportionality than a coarser instrument. For most practical purposes a strain sensitivity of 10^{-5} inch per inch is probably sufficient, and for temperatures up to about 300 degrees C. (570 degrees F.) the rate of loading is not important. Owing to the existence of creep at high temperatures, even at low stresses, the limit of proportionality as measured in a tensile test is a function of both the sensitivity of the extensometer and the rate of loading. During step-by-step loading in a test for the limit of proportionality, creep occurring at a rate of 10^{-5} inch per inch per hour at any particular load would remain undetected during the relatively short time that load was applied, but this rate of creep, if continuous, would result in an extension of about 0.1 per cent in 4 days.

Using extensometers accurate to 10^{-5} inch per inch and a rate of loading of the order of 0.5 ton per square inch per minute, a series of values of the limit of proportionality at various temperatures is obtained which has some practical value at moderate temperatures. For wrought steels the trend of change in the limit of proportionality with temperature is shown in Figure 2. The lower curve indicates the general trend of values obtained under the testing conditions just mentioned, whilst the upper curve relates to values obtained with the use of extensometers accurate to about 10^{-4} inch per inch. It must be stated, however, that the change in limit of proportionality with temperature varies somewhat with different steels, and too free a use must not be made of the curves in Figure 2. The curves do not relate to hardened and tempered steels which have relatively high limits of proportionality at room temperature, but may lose this superiority at high temperatures, as indicated by the data in Table 3.

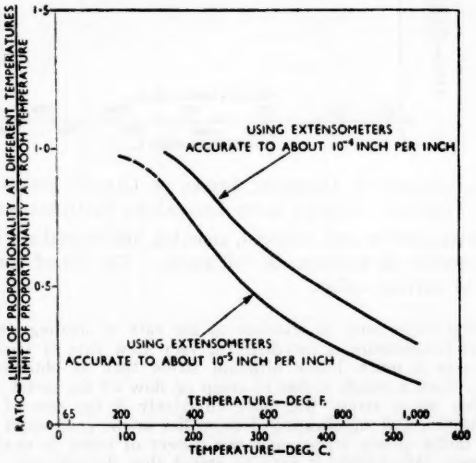


FIGURE 2. CHANGE IN LIMIT OF PROPORTIONALITY OF CARBON STEELS (UP TO 0.6 PER CENT CARBON) WITH INCREASING TEMPERATURE.

The curves refer to cast, wrought, annealed, and normalized steels.

Up to a temperature of about 250 degrees C. (482 degrees F.) the yield point is well defined and its value is not much changed, but beyond about 300 degrees C. (572 degrees F.) no definite yield point exists. As a substitute, a proof stress corresponding to a specified percentage plastic deformation is sometimes determined. When based on a very small plastic deformation its value clearly approaches that of the limit of proportionality. As an independent criterion of the strength of steels at the higher temperatures for the purpose of design, neither the limit of proportionality nor the proof stress serves a useful purpose.

Account has often to be taken of the fatigue properties of steels for use at air or moderately elevated temperatures. Results have been published showing that the resistance to fatigue under cyclic variations of stress taking place at the rate of about 2000 cycles per minute is well maintained

TABLE 3. LIMITS OF PROPORTIONALITY FOR 0·51 PER CENT CARBON STEEL

Steel in two conditions: (*a*) normalized; (*b*) oil-quenched and tempered (600 deg. C.).

Temperature	Normalized condition		Oil-quenched and tempered condition	
	Limit of proportionality, tons per sq. in.	Ratio to value at room temperature, per cent	Limit of proportionality, tons per sq. in.	Ratio to value at room temperature, per cent
Room . .	17·9	100	27·4	100
400 deg. C. (752 deg. F.) . .	7·7	43	10·3	38
450 deg. C. (842 deg. F.) . .	6·2	35	6·4	23
500 deg. C. (932 deg. F.) . .	4·7	26	<4	<15

up to 500 degrees C. (932 degrees F.) At this temperature the ranges of cyclic stress which can be applied to 0.17 per cent carbon steel at 2000 cycles per minute without fracture within 10 million cycles (covering a period of about 3 days) are 11.7 tons per square inch compression to 11.7 tons per square inch tension, zero stress to 17 tons per square inch tension, and 8 tons per square inch tension to 20 tons per square inch tension. These ranges of stress so far exceed those which are likely to be superimposed on a structure at high temperatures in practice that the possibility of fatigue under service conditions can be ruled out.

II. CREEP PROPERTIES.

The subject of creep properties will be considered in three steps. First is given a brief outline of the general creep characteristics of carbon steels, then a description of the extent to which the creep properties have been found to vary amongst the steels, and finally it is shown how existing information may be utilized for design purposes.

(*a*) *General Creep Characteristics.*—Persistence of creep at high temperatures at stresses low in comparison with the tensile ultimate stress is perhaps the most important characteristic from a practical point of view. An illustration may be cited from the results of a creep test on a specimen of 0.13 per cent carbon steel at 450 degrees C. (842 degrees F.) with a stress of 7 tons per square inch; the ultimate stress at 450 degrees C. was about 21.5 tons per square inch. The creep specimen stretched persistently for nearly 3 years and then broke. Over a long period the rate of creep was no greater than about 3 millionths of an inch per inch per day. The long period referred to is known as the second stage of creep during which the rate of creep does not change much with time and passes through its minimum value. The first stage of creep commences at the moment of applying stress and is characterized by a high initial creep rate followed

by a continuous decrease in the creep rate until the second stage is entered. In the third or final stage of creep rate begins to increase again, and when failure is not far off the creep rate may be many hundred times the minimum rate.

In practice it is essential not only to avoid failure of parts under stress but also to avoid undue distortion. Economic considerations lead to choice of working stresses as high as are consistent with a permissible creep deformation, which is usually of so small a value that it is assumed that creep in parts under stress will not enter the third stage, as defined above, during service life. Whether they do so or not depends on the temperature and on the amount of deformation allowed. As regards the deformation, consider the creep curves in Figure 3 for normalized 0.13 per cent carbon steel tested at 450 degrees C. At 10 tons per square inch the minimum creep rate occurs at a deformation of about 1.45 per cent reached in 90 days; at 9 tons per square inch it occurs at about 0.8 per cent creep in 195 days; and at 7 tons per square inch it occurs at about 0.4 per cent creep in 500 days. Judging from the trend of the curve in Figure 3 representing the locus of the points of minimum creep rate, it appears that the deformation corresponding to the minimum creep rate reached in 100,000 hours (11.4 years) would definitely be less than 0.4 per cent. Thus it is evident that a deformation of 0.4 per cent in 100,000 hours is not permissible for this steel at 450 degrees C. if at the same time it is essential that the third stage of creep should not be reached during that period of time.

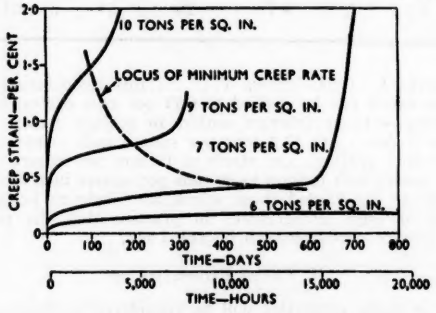


FIGURE 3. LONG TIME CREEP TESTS ON 0.13 PER CENT CARBON STEEL (NORMALIZED) AT 450 DEGREES C.

Considering now the effect of higher temperatures, Figure 4 shows two creep curves for 0.39 per cent carbon forged steel, one for a test at 513 degrees C. (955 degrees F.) at a stress of 5 tons per square inch and the other for a test at 550 degrees C. (1022 degrees F.) at a stress of 3 tons per square inch. It will be seen that the minimum creep rate occurs at a deformation of 0.2 per cent at 513 degrees C. and of 0.16 per cent at 550 degrees C. At lower stresses than those stated in Figure 4, the deformation at the point of minimum creep rate would be lower than the values quoted, so if carbon steels should be used at these temperatures it appears that the permissible deformation must be limited to 0.1 per cent or even less if entry into the third stage of creep is to be avoided. It may be concluded that for the same stress the amount of deformation occurring up to the point of minimum creep rate decreases with increase in temperature; this is to be

expected, since at the higher temperatures little strain hardening of the steels during creep is possible. It is not essential to avoid entry into the third stage, providing that the permissible total deformation is limited to such an amount as to ensure that the rate of creep is not increasing too rapidly towards the end of service life.

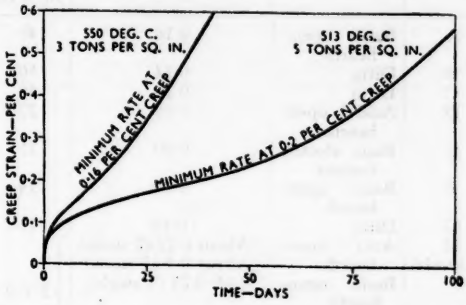


FIGURE 4. DEFORMATION AT POINT OF MINIMUM CREEP RATE FOR 0.39 PER CENT CARBON FORGED STEEL.

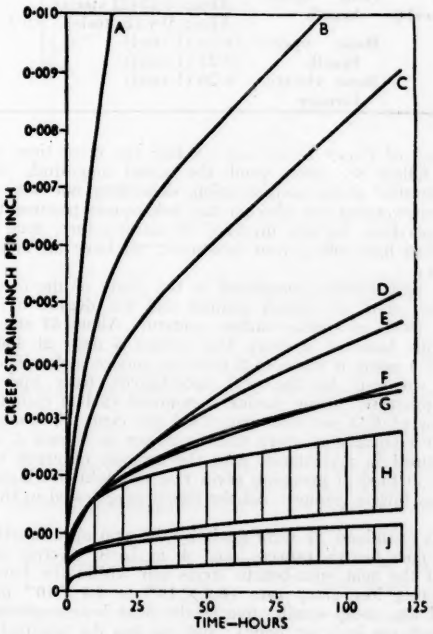


FIGURE 5. SUMMARY OF CREEP RATES, AT 5 DAYS, OF 47 STEELS TESTED AT 450 DEGREES C. UNDER A STRESS OF 8 TONS PER SQUARE INCH.

Curve	Method of manufacture	Carbon content, per cent	Creep rate, in. per in. per hr. $\times 10^{-6}$
A	Basic open-hearth	0·16	140
B	Ditto	0·14	50
C	Ditto	0·20	40
D	Acid open-hearth.	0·19	22·5
E	Basic electric furnace	0·40	21
F	Basic open-hearth	0·19	14
G	Ditto	0·18	11·7
H (12 steels)	Acid open-hearth	About 0·22 (2 steels) About 0·4 (3 steels)	2·3–9·0
	Basic open-hearth	0·15–0·21 (5 steels)	
	Basic electric furnace	0·13 (1 steel) 0·44 (1 steel)	
J (28 steels)	Acid open-hearth	About 0·15 (2 steels) About 0·22 (21 steels) About 0·4 (2 steels)	0·5–2·5
	Basic open-hearth	0·16 (1 steel) 0·25 (1 steel)	
	Basic electric furnace	0·20 (1 steel)	

(*b*) *Variation of Creep Properties.*—While the creep-time curves of all carbon steels follow the same trend, the actual magnitude of creep is a function of the steel under consideration, depending not only on the final product as manufactured but also on any subsequent treatment given to it. Chemical composition, factors involved in manufacture, final form (such as casting, rolled bar, tube), heat treatment, all have an influence on the creep properties.

Two of the first matters considered in the study of the creep of carbon steels were the effect of carbon content and the degree of variation of creep amongst steels of similar carbon content. About 47 steels have been compared on the basis of a creep test lasting 5 days at 450 degrees C. (842 degrees F.) using a stress of 8 tons per square inch. The steels were manufactured variously by the acid open-hearth, basic open-hearth, and basic electric processes, many having a nominal carbon content of 0.2 per cent, and a few of 0.15 per cent and of 0.4 per cent. A summary of their behavior in the comparative creep tests is shown in Figure 5, in which the steels are grouped in accordance with the amount of creep strain during the test. This method of grouping gives rise to slight overlapping of creep rates in the two lowest groups. All the steels were tested in the normalized condition.

Of the steels examined, 31 were made by the acid open-hearth process, 12 by the basic open-hearth process, and 4 in basic electric arc furnaces. Twenty-five of the acid open-hearth steels fall within the lowest group in Figure 5 denoting low creep rate (0.5×10^{-6} to 2.5×10^{-6} inch per inch per hour) and low creep strain; five in the next lowest group (2.3×10^{-6} to 9.0×10^{-6} inch per inch per hour); and one has the relatively high creep

rate of 22.5×10^{-6} inch per inch per hour. Two basic open-hearth steels fall within the lowest group, five in the second group, two more lie somewhat above the second group, and the last three have high creep rates. One basic electric furnace steel has a relatively high creep rate (21×10^{-6} inch per inch per hour); two others fall within the second group; and the fourth is amongst the steels of lowest creep rate. On the basis of this examination it would appear that the acid open-hearth process was more successful than the basic open-hearth process in producing steels of good resistance to creep; but since undertaking these tests a member of the Committee has stated that the cause and cure of the poor creep resistance found in some basic open-hearth steels is now believed to be known.

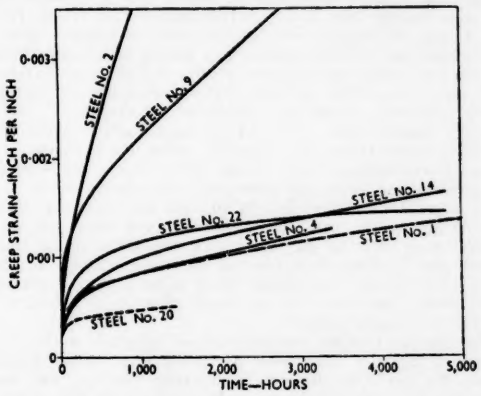


FIGURE 6. LONG TIME CREEP TESTS ON 7 STEELS AT 450 DEGREES C. UNDER A STRESS OF 6 TONS PER SQUARE INCH.

Steel No.	Method of manufacture	Carbon content, per cent	Approximately steady creep rates, in. per in. per hr. $\times 10^{-7}$
2	Basic electric furnace	0·40	—
9	Acid open-hearth	0·42	7·1 between 1,000 and 3,000 hours
4	—	0·37	$1{\cdot}7_5$ between 2,000 and 3,000 hours
14	Acid open-hearth	0·40	$1{\cdot}5_5$ between 2,000 and 5,000 hours
1	Basic electric furnace	0·13	1·0 between 3,000 and 5,000 hours
20	,, ,, ,,	0·20	0·9 at 1,000 hours
22	Acid open-hearth	0·40	$0{\cdot}2_5$ between 3,000 and 5,000 hours

Chemical and metallurgical analyses of several of the steels of both good and poor resistance to creep undertaken with a view to establishing the cause of "abnormal" creep behavior have been discussed elsewhere. Although work of several years ago disclosed the abnormal behavior of some carbon steels, it is possible that "abnormality" may still be encountered until the cause is fully understood. The problem is receiving the closest attention by the Committee, and steelmakers are fully conversant with the subject. In this connection it should be stated that 24 of the acid open-hearth steels included in Figure 5 were samples from recent boiler drum forgings produced by two manufacturers, and in no case was there any abnormal creep behavior.

Although the above comparison of steels has been made on the basis of creep tests lasting only 5 days (120 hours) and at the relatively high stress of 8 tons per square inch, creep tests on selected steels have been made for periods up to about 5000 hours, giving results as shown in Figure 6. The full-line creep curves refer to five 0.4 per cent carbon steels and the dotted curves to a 0.13 per cent carbon and a 0.2 per cent carbon steel, respectively, the tests having been made at 450 degrees C. and at a stress of 6 tons per square inch. The 0.4 per cent carbon steels Nos. 2 and 9 show a much faster creep rate than the other 0.4 per cent carbon steels. Of the two lower-carbon steels chosen for comparison, steel No. 1 (0.13 per cent carbon) has about the same creep resistance as the higher-carbon steels Nos. 4 and 14, but steel No. 20 (0.2 per cent carbon) shows distinctly less creep strain in the first few hundred hours than any of the others. The relative creep rates of the steels vary somewhat as the time of testing is increased, and further divergence of the creep curves for the group of steels Nos. 1, 4, 14, and 22 appears likely after a longer period of testing than 5000 hours, but there is no doubt that steels Nos. 2 and 9 are distinctly inferior to the others.

Some interesting further comparisons of selected steels are shown in Table 4, which includes the results of tensile tests at air temperature and 450 degrees C. and data derived from creep tests at 400, 450, and 550 degrees C. for some of the lower-carbon steels. In the first place, it is evident from Table 4 that the tensile properties at air temperature give no indication of the relative creep properties as judged by the results of 20 days' tests at 450 degrees C. (last column). Tensile tests at 450 degrees C. at a pulling speed of 0.001 inch per inch per minute give lower values of ultimate stress for the steels of poor creep resistance, but the drop in ultimate stress is not much more than the corresponding differences in ultimate stress at air temperature. No definite correlation of creep properties with the limit of proportionality at 450 degrees C. is evident. Perhaps the most significant factor in the tensile tests at 450 degrees C. is the apparent correlation between increase in ductility (percentage elongation) and decrease in creep resistance at 450 degrees C.

In view of the wide range in the creep rates of the steels, it was thought likely that the quality of a steel might be disclosed by a short-time creep test. One test method consisted in determining the stress corresponding to 1 per cent creep in 24 hours at 550 degrees C. from the results of two or three tests at different stresses. The results obtained for four of the lower carbon steels are given in Table 4 (col. 10). If it were specified that a stress of at least 6 tons per square inch should be attained in this test for a steel of requisite quality, the two steels (No. 15 and P) of poor creep resistance would be eliminated. Another method consisted in determining the creep rates of the steels at 450 degrees C. after 2 hours under a stress of 6 tons per square inch. The results recorded in Table 4 (col. 11) show that these single creep tests also served to separate the good from the poor-quality steels. Table 4 also contains the results of a few creep tests

TABLE 4. COMPARISON OF TE

Steel No.	Carbon content, per cent	Tensile tests temperatu	
		Limit of proportionality, tons per sq. in.	Yield point, tons per sq. in.
1	2	3	4
20	0·20	20·2	22·0
19	0·16	11·1	15·8
21	0·15	19·8	19·8
1	0·13	16·5	17·1
15	0·19	8·9	14·4
P	0·16	20·4	20·4
10	0·40	13·6	19·1
16	0·44	22·8	23·4
14	0·40	20·7	22·2
9	0·42	10·0	—
2	0·40	18·5	23·8

Steel No.	Type	
		Carb
20	Basic electric	0·2
19	Acid open-hearth	0·1
21	" "	0·1
1	Basic electric	0·1
15	Acid open-hearth	0·1
P	Basic open-hearth	0·1
10	Acid open-hearth	0·4
16	Basic electric	0·4
14	Acid open-hearth	0·4
9	Acid open-hearth	0·4
2	Basic electric	0·4

SON OF TENSILE AND CREEP PROPERTIES OF "NORMAL" AND "ABNORMAL" CARBON STEELS

Tensile tests at air temperature			Tensile tests at 450 deg. C. (strain rate 0·001 in. per in. per min.)			Creep tests at 400 deg. C. (stress 15 tons per sq. in.)	Creep tests at 550 deg. C.	Creep tests at 450 deg. C. (stress 6 tons per sq. in.)	Creep tests at 450 deg. C. (stress 8 tons per sq. in.)
t of or-ity, per n.	Yield point, tons per sq. in.	Ultimate stress, tons per sq. in.	Limit of proportionality, tons per sq. in.	Ultimate stress, tons per sq. in.	Elongation at fracture (gauge length = $4\sqrt{\text{area}}$), per cent	Creep rate at 15 days, in. per in. per hr. $\times 10^6$	Stress for 1 per cent total strain in 24 hours, tons per sq. in.	Creep rate at 2 hours, in. per in. per hr. $\times 10^6$	Creep rate at 20 days, in. per in. per hr. $\times 10^6$
	4	5	6	7	8	9	10	11	12
2	22·0	31·9	2·8	20·4	36	4·2	$6·3_5$	9	0·3
1	15·8	29·0	—	—	—	—	—	—	0·8
3	19·8	30·9	4·0	21·1	40	—	$6·7_5$	10	0·8
5	17·1	27·2	—	—	—	—	—	—	1·3
9	14·4	27·8	2·4	17·7	54	71	4·6	80	10·8
4	20·4	27·1	2·0	15·6	63	67	$3·8_5$	206	107·0
6	19·1	39·4	—	24·5	41	—	—	—	1·5
8	23·4	40·3	—	—	—	—	—	—	1·5
7	22·2	39·0	—	—	—	—	—	—	1·7
0	—	38·3	—	22·6	45·5	—	—	—	2·1
5	23·8	37·2	—	20·8	49	—	—	—	18·5

	Chemical composition, per cent							
	Carbon	Silicon	Sulphur	Phosphorus	Manganese	Nickel	Chromium	Copper
	0·20	0·20	0·023	0·011	0·56	0·162	Trace	0·10
rth	0·16	0·14	0·037	0·034	0·66			
	0·15	0·28	0·028	0·030	0·76	0·091	Trace	0·11
	0·13	0·16	0·038	0·029	0·53	0·03	Trace	0·08
rth	0·19	0·08	0·040	0·051	0·40			
rth	0·16	0·006	0·032	0·028	0·46			
rth	0·40	0·16	0·041	0·042	0·53	Trace	Trace	0·12
	0·44	0·14	0·026	0·011	0·55	0·05	0·04	0·05
rth	0·40	0·21	0·037	0·035	0·58			
rth	0·42	0·31	0·031	0·031	0·50	0·09	Trace	0·06
	0·40	0·21	0·038	0·033	0·59	0·03	Trace	0·08

at 400 degrees C. (col. 9); these results, together with those at 450 degrees C. and 550 degrees C. show that the poor creep resistance of certain of the steels is not confined to one temperature.

(*c*) *Utilization of Data for Design Purposes.*—Many authorities consider that the working stress should be a function of a predetermined stress estimated to produce not more than the permissible deformation by creep during service life. In certain cases, as in steam power plant, the service life is reckoned as being of the order of 100,000 hours (11.4 years), whilst the permissible deformation may be as low as 0.01 per cent in the case of turbine disks or as much as 0.3 per cent in the case of steam pipe work. Deducting working stresses on the above basis appears to be the most rational method, but it is not a simple matter to provide the basic experimental data. It has already been shown to what extent the creep rates of steels of similar composition may vary, and Figure 6 illustrates an important factor, namely, the varying amount of initial creep amongst steels which, after several thousands of hours under the same stress, may settle down to about the same creep rate. From tests of 1000 hours' duration it is not possible to predict with sufficient accuracy the relative creep deformations of a number of steels after 5, 20, or 100 times the testing period.

By measuring, for a given steel, the actual times to produce specified amounts of creep under different stresses from tests extending to at least 10,000 hours, precise data are obtainable over a useful period of time, and the results obtained by extrapolating the data to 100,000 hours are more reliable than estimates from tests of 1000 hours' duration. Even so, the actual deformations at 100,000 hours of these apparently similar steels under the stress estimated to produce 0.1 per cent strain may be expected to vary both below and above 0.1 per cent. When it is considered that good-quality steels of between 0.15 and about 0.5 per cent carbon content appear to have much the same creep properties, and that the creep rate and total deformation during service are much influenced by inconstancy of service temperature, it seems sufficient for design purposes to provide reliable creep

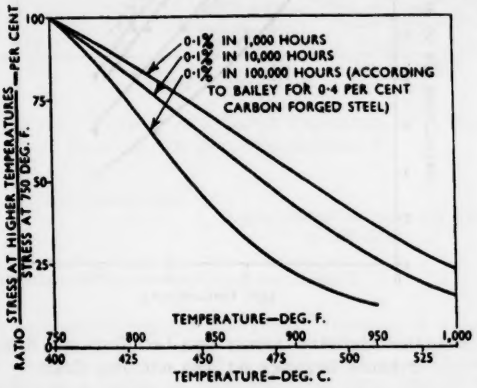


FIGURE 7. DECREASE OF THE STRESS CORRESPONDING TO 0.1 PER CENT CREEP IN SPECIFIC TIMES FOR TEMPERATURES ABOVE 750 DEGREES C.

Summarized from published information.

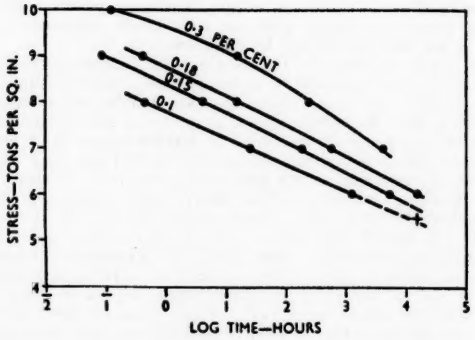


FIGURE 8. RELATION BETWEEN STRESS AND LOG TIME FOR SPECIFIED CREEP STRAINS BETWEEN 0.1 AND 0.3 PER CENT.

+ Estimated value.

Results refer to 0.13 per cent carbon steel of average long-time creep properties. Temperature, 450 degrees C.

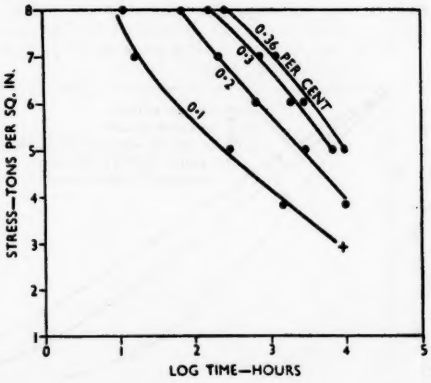


FIGURE 9. RELATION BETWEEN STRESS AND LOG TIME FOR SPECIFIED CREEP STRAINS BETWEEN 0.1 AND 0.36 PER CENT.

+ Estimated value.

Results refer to 0.42 per cent carbon steel of poor long-time creep properties. Temperature, 450 degrees C.

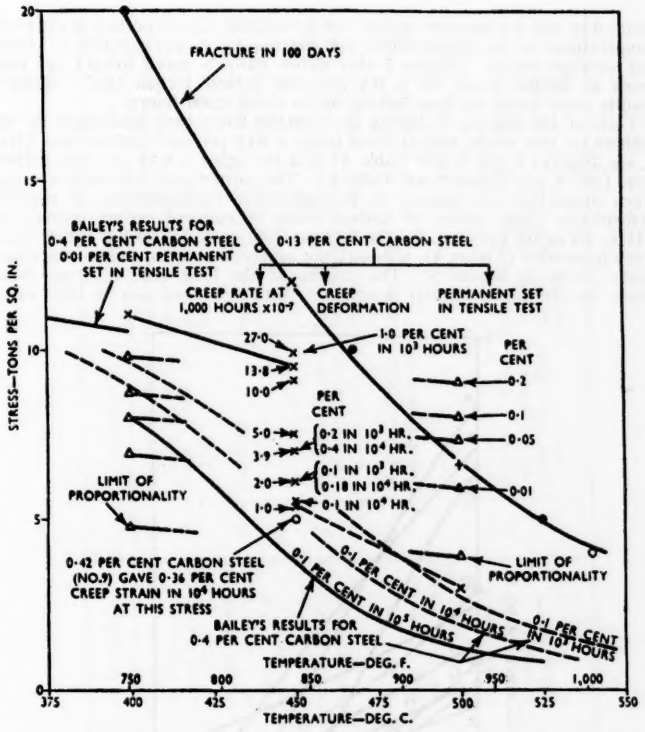


FIGURE 10. RELATED TENSILE AND CREEP PROPERTIES OF CARBON STEEL BETWEEN 750 AND 1000 DEGREES F.

+ 0·13 per cent carbon steel
○ 0·17 ,, ,, ,,
● 0·4 ,, ,, ,,
+ 0·51 ,, ,, ,,

} Points plotted in curve for fracture in 100 days.

For particulars of the materials, etc., see Table 5.

data for one representative steel of the group. However, it is necessary to take into account the lower strength of the steels when in the spheroidized condition as compared with the hot-rolled or normalized conditions.

As the result of a review of published creep data, two of the curves in Figure 7 have been prepared. They represent the average decrease in stress with increase in temperature beyond 750 degrees F. (399 degrees C.) for a deformation of 0.1 per cent in 1000 and 10,000 hours. The data collected for the preparation of the two curves relate mainly to tests not exceeding 1000 hours' duration and to steels of various compositions between

about 0.15 and 0.5 per cent carbon, but by relating the properties at different temperatures to one temperature, 750 degrees F., it was possible to obtain fair average curves. Figure 7 also shows Bailey's curve for 0.1 per cent creep in 100,000 hours for a 0.4 per cent carbon forged steel. Bailey's results were based on tests lasting up to about 2,000 hours.

Tests at 450 degrees C. lasting up to 20,000 hours have been made by the authors on two steels, one of these being a 0.13 per cent carbon steel (No. 1, see Figures 3 and 6 and Table 4) and the other a 0.42 per cent carbon steel (No. 9, see Figure 6 and Table 4.) The former steel has fairly average creep properties and appears to be sufficiently representative, as regards deformation under stress, of carbon steels of nominal carbon content of 0.15 to 0.4 or 0.5 per cent, but the 0.42 per cent carbon steel has rather poor creep properties (Figure 6), although not nearly so poor as some of the other steels shown in Figure 5. The results of the long time tests on these steels are shown in Figures 8 and 9. With one exception in each case,

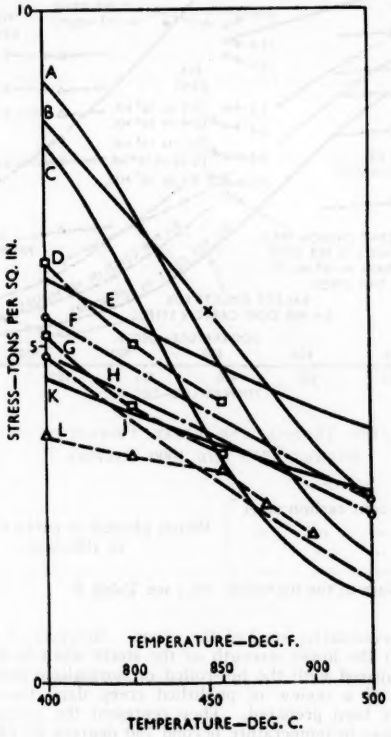


FIGURE 11. PROPERTIES OF CARBON STEEL BETWEEN 400 AND 500 DEGREES C., AND SOME PUBLISHED VALUES OF WORKING STRESSES.

TABLE 5

Curve	Source	Material	Remarks
A	Bailey	0·4 per cent carbon forged steel	0·1 per cent creep in 10^4 hours
B	National Physical Laboratory	0·13 per cent carbon steel; tensile strength, 27 tons per sq. in.	0·1 per cent creep in 10^4 hours (curve based on an equivalent curve in Fig. 7, p. 104)
C	Bailey	0·4 per cent carbon forged steel	0·1 per cent creep in 10^5 hours
D	American Standards Association (B31–1–1935)	Steel for piping; tensile strength, 33·5 tons per sq. in.	Working stress
E	National Physical Laboratory	Carbon steel	Limit of proportionality for steel having value of 20 tons per sq. in. at air temperature
F	Hatfield	Normalized bar; 0·25 per cent carbon	Two-thirds of "time yield"
G	American Standards Association (B31–1–1935)	Steel for piping; tensile strength, 27·5 tons per sq. in.	Working stress
H	Hatfield	Normalized forging; 0·24 per cent carbon	Two-thirds of "time yield"
J	National Physical Laboratory	Normalized carbon steels; 0·2–0·4 per cent carbon	One-third limiting creep stress (Tapsell)
K	National Physical Laboratory	Carbon steel	Limit of proportionality for steel having value of 15 tons per sq. in. at air temperature
L	British Standard Specification No. 806 (1938)	Steel for piping; tensile strength, 22–30 tons per sq. in.	Working stress

all the plotted points are actual experimental determinations. The curves show that the divergence in resistance to creep of the two steels increases as the stress is reduced from 8 tons per square inch, and that the stresses corresponding to 0.1 per cent deformation in 10,000 hours at 450 degrees C. are approximately 5.5 tons per square inch for the 0.13 per cent carbon steel and 3.0 tons per square inch for the 0.42 per cent carbon steel. These stresses are respectively above and below the value (about 5.05 tons per square inch) found by Bailey for 0.4 per cent carbon forged steel. Considered from the point of view of relative deformation, the 0.42 per cent carbon steel deformed 0.55 per cent in 10,000 hours at 5.5 tons per square inch compared with a deformation of 0.1 per cent for the 0.13 per cent carbon steel. Although some of the curves in Figures 8 and 9 extend to a period of 10,000 hours, the authors are not prepared to maintain that linear or roughly linear extrapolation to 100,000 hours will necessarily lead to a precise estimate of the stress for a specified deformation in 100,000 hours.

Some of the results of the long-time creep tests on the 0.13 per cent carbon steel are incorporated in Figure 10, which includes limits of proportionality and various proof stresses for this steel, together with Bailey's results for 0.4 per cent carbon forged steel. In the case of the latter steel, the limit of proportionality and the proof stresses (not shown) are higher than those for the 0.13 per cent carbon steel. Bailey's curve for 0.1 per cent creep in 1000 hours lies just above the value obtained for the 0.13 per cent carbon steel at 450 degrees C., but his value for 0.1 per cent creep in 10,000 hours is about 5 tons per square inch against the value of 5.5 tons per square inch for the 0.13 per cent carbon steel. Figure 6 shows that some 0.4 per cent carbon steels as well as a 0.20 per cent carbon steel are equally as good at 450 degrees C. as the 0.13 per cent carbon steel; and two other 0.4 per cent carbon steels are much inferior. At 400 degrees C. (Figure 10) Bailey's curves for 0.1 per cent in 10,000 and 1000 hours both extend beyond the stress corresponding to a tensile permanent set of 0.1 per cent for the 0.13 per cent carbon steel, so they are not applicable to the 0.13 per cent carbon steel at that temperature.

In a final summary of the creep properties of carbon steels in relation to design stresses, the attitude taken is that all good-quality carbon steels of about 0.15-0.5 per cent carbon content, in the hot-rolled or normalized conditions, may, for practical purposes, be considered to have similar creep properties between 400 and 510 degrees C (750 and 950 degrees F.) but that, in view of their different hot-tensile properties account should be taken of the limits of proportionality (or alternatively the 0.01 per cent proof stresses) up to 426 degrees C. or 455 degrees C. (800 degrees F. or 850 degrees F.). Although the complete curves between 400 degrees C. and 510 degrees C. (750 degrees F. and 950 degrees F.) for 0.1 per cent deformation in 10,000 and 100,000 hours of a representative carbon steel have not yet been determined, it is considered that the information obtained in regard to the 0.13 per cent carbon steel, together with Bailey's results for the 0.4 per cent carbon forged steel, and the slopes of the average curves in Figure 7, taken together, provide sufficient data on which to base working stresses. These data are collected in Figure 11 and Table 5, which includes some published values of working stresses, and also includes two curves representing the limits of proportionality of steels whose equivalent limits at room temperature are 20 and 15 tons per square inch respectively. The latter curves are deducted from the lower curve of Figure 2. The curve for 0.1 per cent deformation in 10,000 hours for the 0.13 per cent carbon steel is based on the experimental result at 450 degrees C. and on the equivalent curve in Figure 7. Many years ago one of the authors suggested that the curve (Figure 11) representing one-third of the limiting creep stress probably formed a satisfactory basis for design, the limiting creep stress being defined

TABLE 6. COMPARISON OF CAST AND FORGED STEELS

Steel	Limiting creep stresses, based on a creep rate of 10^{-5} in. per in. per day at 40 days
0·39 per cent carbon forged steel, annealed and reheated to 550 deg. C. (1,132 deg. F.)	13 tons per sq. in. at 400 deg. C. (750 deg. F.) 10 tons per sq. in. at 447 deg. C. (835 deg. F.) 5 tons per sq. in. at 507 deg. C. (945 deg. F.)
0·30 per cent carbon cast steel, annealed and reheated to 550 deg. C. (1,132 deg. F.)	10 tons per sq. in. at 400 deg. C. (750 deg. F.) 8 tons per sq. in. at 450 deg. C. (840 deg. F.) 4 tons per sq. in. at 497 deg. C. (925 deg. F.)
0·53 per cent carbon cast steel (British Standard Specification No. 30, grade A)	10 tons per sq. in. at 415 deg. C. (780 deg. F.)

as the stress corresponding to a creep rate of 10^{-5} inch per inch per day at 40 days, approximately 4×10^{-7} inch per inch per hour at 1000 hours.

If a steel is in a highly spheroidized state it is considerably weaker than when normalized or when air-cooled after hot rolling, and the use of such a steel should be avoided. Slight cold work is not likely to alter the properties of a steel sufficiently to put the steel outside the range of steels of average quality, but heavy cold work would render the steel subject to more rapid spheroidization during service especially in the higher working temperature range. The creep properties of a range of cast steels have not been explored, but the results of creep tests on two cast steels and a forged steel given in Table 6 are probably indicative of the relative strengths of cast and forged steels.

The authors desire to state in conclusion that they have endeavored to set forth their most recent information regarding the creep properties of carbon steels, but they intentionally leave the engineer to decide for himself the design stress suitable to his requirements after satisfying himself that the steel he wishes to use is of "good quality."

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

ARC WELDING WITHOUT RESIDUAL SHRINKAGE STRESSES.—Ralph E. Spaulding, of the Aetna Steel Construction Company, Jacksonville, describes a patented method of eliminating residual shrinkage stresses in this article reprinted from the April, 1941, issue of the Welding Journal.

It is well known that in the ordinary process of arc welding certain shrinkage stresses are set up (sometimes referred to as locked-in stresses) which cause shrinkage along the weld with consequent bending or warpage of the welded assembly. The difficulties encountered because of this are well known.

Before we can take steps to prevent the occurrence of shrinkage stresses due to welding it is necessary to know exactly what physical process causes the shrinkage.

In the welding process the application of intense heat to a localized area causes that highly heated area to expand. This localized area cannot expand in a normal manner, that is, in all directions in accordance with the temperature change times its coefficient of thermal expansion, being restrained by the larger areas of relatively colder metal surrounding the smaller area of hot metal (usually on all sides except the top), and therefore elastic deformation, or plastic deformation, takes place in the highly heated area along the line of least resistance. The fact that the modulus of elasticity as well as the yield strength of highly heated metal is relatively low makes this deformation comparatively easy.

To illustrate this assume a bar, as in Figure 1, and lay a welding bead throughout its length along its center line, y-y.

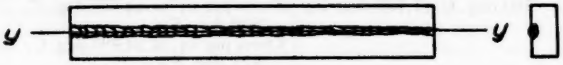


FIGURE 1.

At any cross section of the bar at the moment the weld bead has just been laid we have the condition described above, that is, an elongated spot of highly heated metal which has undergone plastic deformation, its high relative temperature having caused it to expand. Because of the vastly greater strength of the whole bar at normal temperature as against the relative softness and weakness of the hot spot, this highly heated spot cannot expand along the axis of the bar y-y but can, and does, expand and plastically deform outwardly and away from the mass of the bar. Therefore, taking the entire bead along the axis y-y into consideration it is evident that the bead, together with the adjacent metal which has been highly heated, has an absolute axial length along the axis y-y, *in its highly heated state*, which is the same as the axial length of the bar itself *at its relatively colder temperature before welding.*

As the entire bar plus its deposited weld bead again becomes of one uniform temperature this formerly highly heated area obviously undertakes to contract in all directions, including along the axis y-y, in accordance with its coefficient of thermal contraction. While relatively free to contract inwardly, that is, from the face toward the root of the weld, it is restrained from contracting in *all* directions, particularly along the axis y-y, by the superior bulk of the surrounding cooler metal so that the weld area (comprising the weld bead plus the highly heated parent metal) is subjected to tension along the axis y-y.

To more fully describe this physical process we refer to Figure 2, which is a graph of temperature change and internal pressure of the spot undergoing heating and subsequent cooling as outlined above.

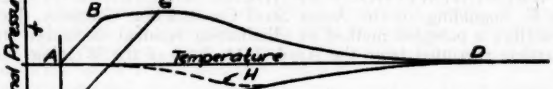


FIGURE 2.

Referring to Figure 2, the horizontal line running through A-D represents the measure of change in temperature, and the point D on this line the highest temperature to which the spot is subjected. Vertical distances above the horizontal temperature line represent the measure of internal pressure produced by the change in temperature, and distances below the horizontal line represent the measure of internal negative pressure—or tension—in the spot undergoing temperature change.

For corresponding unit areas let t denote change in temperature; d, deformation; E, modulus of elasticity; Y, elastic limit, or yield strength; K, coefficient of thermal expansion; f, unit fiber stress (or pressure).

Then, starting at room temperature A, as heat is applied to the spot the metal expands in all directions where free to do so in accordance with the well-known formula

$$d = tK$$

As previously stated it is not free to expand in all directions, particularly not along the axis y-y. In the direction in which expansion is restrained an internal pressure is built up in accordance with the formula

$$f = dE = tKE$$

Plastic deformation cannot take place until the internal stress has become as great as the yield strength of the metal, or until

$$f = Y$$

This point is denoted by B in the diagram (Figure 2). At the point B no plastic deformation has as yet taken place but the internal stress is just equal to the yield strength. If, then, the spot be allowed to cool again to room temperature, without further heating, the internal pressure would decrease back down to zero at A, and there would remain no locked-in stress. If, however, heat continues to be added beyond point B, plastic deformation will take place and continue so long as heat is added, the internal stress always being equal to the yield strength, increasing as yield strength increases and decreasing to zero as the metal reaches the melting point, at D.

As the temperature starts (from D) reducing back to normal a condition the reverse of the above takes place, the metal becomes subjected to negative pressure, or tension, in accordance with the same law of $f = tKE$ as above. This tension increases until it equals the yield strength of the metal (at the then instant temperature) whereupon plastic deformation again begins to take place, and plastic deformation continues so long as the temperature continues to drop, the internal tension always being equal to the then instant yield strength, until, when the metal has cooled back to its original starting temperature, F, the internal tension, particularly along axis y-y, is equal to the full yield strength of the metal.

The temperature change necessary, in most structural steels, to produce an internal pressure equal to the yield strength, or $t = Y/KE$, is approximately 200 degrees F.

All metal which is adjacent to the weld but not subjected to an increased temperature of more than approximately 200 degrees F. returns to its normal state upon cooling without locked-in stresses, its temperature-pressure curve being A to B and B back to A.

But if a spot be heated to a maximum of twice 200, or approximately 400 degrees F. (represented by point G on the curve), its temperature-pressure curve during the complete cycle of heating and cooling is: A to B at which point plastic deformation starts, B to G during which period plastic deformation is taking place, and the cooling period of G to F during which no plastic deformation takes place but the internal stress changes from full yield strength in compression at G to full yield strength in tension at F, where the temperature is again normal.

Therefore all metal of the weld area in which the change in temperature has not exceeded approximately 200 degrees F. has no locked-in stresses.

Metal in which the temperature rise has ranged from approximately 200 degrees F. to approximately 400 degrees F. contains locked-in stresses varying from zero to full yield strength.

All metal in which the temperature rise has exceeded approximately 400 degrees F. contains after the weld has cooled, locked-in stresses equal to the full yield strength of the metal.

This tension is balanced by compression in the remainder of the metal with resultant shrinkage and warpage.

All welding shops are familiar with this shrinkage and have their own methods of counteracting the warpage, usually by producing, by applied heat, another zone of shrinkage on the opposite side of the center of gravity of the piece (heat balancing) leaving the welded assembly slightly shorter than the original length, and leaving in the assembly two sets of locked-in stresses instead of one.

The locked-in stresses can be relieved in an annealing oven, but it is expensive and usually does not restore the warped assembly to its original shape. Furthermore, in a great majority of cases it is practically impossible to apply this method.

It is, however, possible and comparatively easy and simple to prevent the initial development of this shrinkage stress in the weld area.

As has been above described, the weld area (consisting of deposited weld metal plus all of the adjacent original metal which has been heated to at least approximately 400 degrees F. above its original temperature) is, while cooling, trying to adapt its normal axial length to that of the surrounding original metal but cannot do so by itself alone. If only a very little assistance is given during this period of cooling the axial length of the weld area can readily adapt itself to the normal axial length of the original metal. Obviously this assistance must take place during the initial cooling of the weld area.

It can be accomplished by pressing or light hammering of the weld area while it is cooling (vibratory pressure is best or pressure only if welding process is continuing). This added pressure, applied at right angles to the axis of the weld, can usually be best accomplished by the use of a light pneumatic or electric hammer having a tool face adapted to the desired size and contour of the weld. If applied at the right time, that is, while the yield strength of the hot weld is relatively low, only a small amount of effort is needed and all shrinkage tendency is found to have been eliminated. The temperature-pressure curve during this operation is illustrated by the dotted line H, Figure 2.

Our shop has used this method* on certain welded assemblies for over a year with perfect results.

A test was made under the supervision of the Pittsburgh Testing Laboratories and their report shows:

(1) That the tendency to shrinkage is eliminated.

(2) That the weld when treated as described is stronger than an equivalent untreated weld.

This method must not be confused with so-called "peaning" or hammering of the weld area after it has cooled to such an extent that the elastic limit, or yield strength, of the area has again approximated that of the original metal. Nor should it be compared to reheating of the weld area, with or without treatment while heated, since no heat produced by a flame can even approximate the same relative heat penetration with corresponding surface intensity

* The Aetna Iron & Steel Co. has patents covering this above-described method.

of that resulting from an electric arc. In both of these latter methods counterbalancing locked-in stresses may be introduced which will overcome warpage but the welded assembly is nevertheless still subjected to high internal stresses, and often the physical structure of the metal is disturbed and weakened.

PIONEERING IN WIND POWER.—Recent years have seen various attempts to revive the "windmill" as a modern well-engineered wind turbine, but nothing on the scale or of the quality described in this article reprinted from the June, 1941, issue of Power. In an age when the wind has been relegated to minor farm chores while all major power demands are served by steam, hydro and internal-combustion engines, this plant will, according to Power's editorial, attract wide attention.

Windmills have served man for many centuries as small power units to grind his grain, pump his water and, in more recent times, to generate his electric power. Outside of a change from wood to steel construction they have remained practically the same old windmills of song and story. Recent developments in aerodynamics, metallurgy, aerology and related technology have made possible the production of the Smith-Putnam wind turbine, invented by Palmer Cosslett Putnam, a Boston engineer, which gives promise of economic large-scale power generation from the wind.

This work has progressed so far that in a few weeks the largest aeroelectric-power-generating unit ever built and the first to supply alternating current for commercial use, will go into service. The installation is being made on top of a 2000-foot mountain, Grandpa's Knob, near Rutland, Vt., and will tie into the power system of the Central Vermont Public Service Corp.

Built at a cost of several hundred thousand dollars, this installation combines one of the largest industrial research projects ever attempted with a 1000-Kw. wind-power plant. The research work is divided into (1) an exhaustive study of winds, and (2) of wind-power-plant performance to provide the fundamental and engineering knowledge necessary for economic design, construction and operation of such plants. After the research work is completed there will remain an aeroelectric plant to produce power for the local utility system.

The entire wind-power plant is balanced on top of a structural steel tower 110 feet high, 36 feet square at the base and anchored into the mountain top with a steel grillage and concrete foundation 23 feet deep. This tower was built by the American Bridge Co. and near it a 180-foot anemometer tower has been erected for measuring and recording wind velocities during the experimental period.

Anemometer Installation.

On the anemometer tower are 19 instruments of three different designs, placed on long spars at several levels. Some of the anemometers are gas heated so that wind velocities may be studied during the severest ice storms. Each anemometer connects to an automatic recorder in a wind-research laboratory at the foot of the tower. This laboratory is also equipped with automatic wind-direction recorders and equipment for control of the heating of the anemometers.

A heavy plate-steel-girder structure, called the pintle girder, forms the foundation for the plant. It inclines downward toward the wind at an angle of 12.5 degrees so that the turbine shaft makes a similar angle to the horizontal. The pintle girder, the heaviest single part of the plant, is

supported at A on the top end of a pintle shaft. This shaft has a double roller guide bearing just below the girder at B and a combination roller guide and thrust bearing at C, which leaves the whole plant free to yaw with the wind.

The blades and their control mechanism overhang one side of the tower and approximately balance the rest of the equipment on the other side. The blades, constructed of shot-welded stainless steel by the E. G. Budd Mfg. Co., are over 11 feet wide by 65 feet long and weigh 15,300 pounds each. Supporting the blades from the end of the shaft presented many difficult design problems. The supporting structure not only has to cantilever the weight of the blades into the shaft, but weight of the blades may double during an ice storm. Furthermore, the ice may heavily unbalance the two blades. Heavy centrifugal forces must also be taken into account, in addition to various aerodynamic forces produced by winds up to 140 miles per hour.

A heavy structural steel hub-post about 16 feet long projects at right angles to the main shaft and is securely attached to it by a central hub. Each blade is supported by its shank from the hub-post by a heavy structural-steel A-frame hinge. Structural-steel coning struts tie the outer ends of the A-frames to one end of crank arms pivoted in the downward end of the hub, which extends out about 10 feet from the hub-post. The other end of the crank arms connect to hydraulic-cylinder pistons that act as oil dashpots to stabilize the coning of the blades under the impact from gusts of wind.

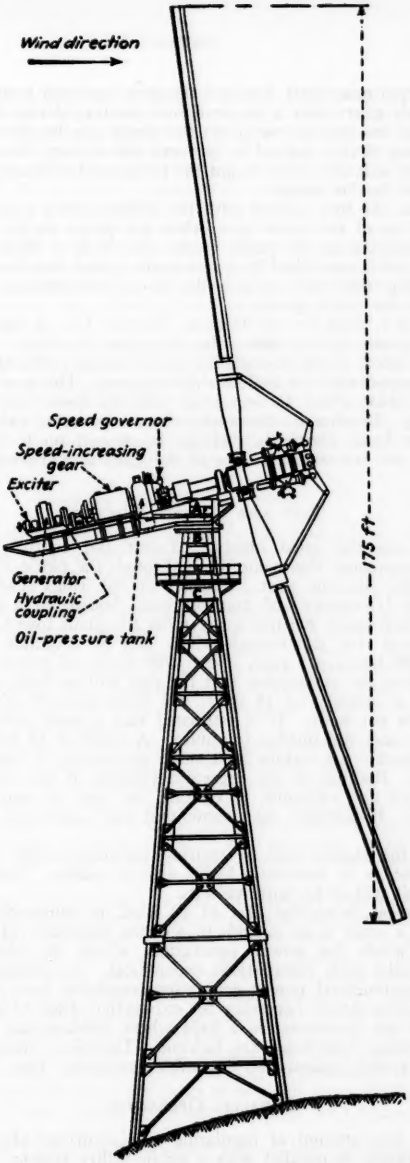
Two heavy roller bearings support each blade shank—one a combination radial and thrust design in the outer end of its A-frame support and the other a radial type in the hub. At their inboard ends, the blade shanks connect by short cranks and linkages to a hydraulic piston in the shaft in such a way that the pitch angle of the blades can be changed through an angle of nearly 100 degrees. Pitch of the blades is controlled in very much the same way as blade pitch is regulated in a Smith-Kaplan hydraulic turbine.

Turbine-Blade Motions.

The blades therefore have three motions: First, they are driven by the wind to rotate the main shaft at 28.7 RPM. Since the blades span 175 feet, a speed of 28.7 RPM. gives them a tip speed of 15,785 feet per minute compared to around 6000 feet per minute for a propeller type hydraulic turbine.

When the blades are being driven at normal speed, centrifugal force holds them at nearly right angles to the main shaft. If a sudden gust of wind hits them they can cone down-wind to a maximum of 20 degrees to relieve the initial shock and assist the governor in regulating speed. As previously mentioned, speed is regulated by the governor changing the pitch angle of the blades into the wind. Change in blade-pitch angle for normal operation will be about 30 degrees, but a range of 99 degrees has been provided for research purposes.

Two double roller bearings 12 feet apart on the pintle girder support the turbine shaft, which is about 24 inches in diameter. The bearing just back of the turbine hub is a radial unit and that at the coupling for the speed-up gears takes both axial thrust and radial loads. The turbine shaft couples to a double-spur-and-herringbone-gear unit that increases its speed to 600 RPM., that of the generator. First step of speed increase is taken by a spur gear meshing into a pinion on either side of it in the horizontal plane. These pinions are on the shaft of two herringbone gears that mesh into the pinion on the output shaft. This shaft connects through an American Blower hydraulic coupling to the generator shaft. Beyond the generator is a direct-connected exciter. A standard Woodward speed governor with some electrical contacts added is gear driven from the turbine shaft.



THE ENTIRE PLANT IS BUILT ON A HEAVY STRUCTURAL-STEEL PINTLE GIRDER ON TOP OF A 110-FOOT TOWER ANCHORED INTO THE MOUNTAIN TOP BY A HEAVY STEEL AND CONCRETE FOUNDATION.

On the output-gear shaft, between the gear case and hydraulic coupling a friction clutch gears into a motor-driven turning device equipped with a brake. When the unit is shut down the clutch can be closed and the brake on the turning device applied to prevent the turbine from turning. The turning device will also serve to put the turbine rotor in any desired position when the unit is shut down.

At all times the unit is held with the turbine down wind by a bull gear fixed in the top of the tower just below the pintle girder. The bull gear connects to gearing on the pintle girder, driven by a Waterbury hydraulic transmission and is controlled by a yaw meter (wind-direction vane) mounted on the housing that will enclose the power-transmitting and generating equipment on the pintle girder.

The generator, built by the General Electric Co., is rated 1000 Kw. at 80 per cent power factor, 2300 volts, 60 cycles, 3 phase. Leads from the generator are taken down through the center of the pintle shaft and connect to slip rings supported on a hollow slip-ring post. This post is an extension of the pintle shaft about 10 feet long, with its lower end supported in a guide bearing. Brushes on these slip rings connect to cables that lead to a transformer bank where the voltage is stepped up to 44,000 volts for connection to the transmission lines of the Central Vermont Public Service Corp.

Test and Control Circuits.

Conductors for the great number of test and control circuits on the power-plant equipment also come down through the pintle shaft and connect to rings on the slip-ring post. Brushes on the rings lead to the power-plant research laboratory and control room, housed in a heavy steel and concrete building about 200 feet away from the plant tower.

It is expected that satisfactory winds will be available to operate the unit about 4000 hours per year, about 2700 hours of which will be at full load. This is on the assumption that the unit will go into service when the wind reaches a velocity of 18 miles per hour and off at wind velocities above 60 miles per hour. It is estimated that a wind velocity of 4 miles an hour will cause the turbine to rotate. A wind of 18 miles per hour is expected to operate the turbine at a small percentage of rated capacity and normal speed. Because of the uncertain velocity of the wind it is at this time considered not advisable to operate the unit at much less than 25 per cent load. Experience may show that safe operation can be had at lower loads.

Control of the station will be entirely automatic using very much the same equipment as in automatic hydro electric plants. Starting and stopping of the unit will be by wind velocity.

The installation, being the first of its kind, is admittedly experimental. It is built on a scale large enough to test the feasibility of such units for utilizing the winds for power generation, where the annual winds are sufficient to make such installations economical. In addition to designing the unit for commercial power generation, engineers here incorporated in it practically every device necessary for exhaustive study of its performance. The designers are pioneering in a field where fundamental knowledge and practical operating experience are lacking. Therefore, when they were in doubt they provided against every possible emergency they could think of.

Parallel Operation.

This is the first attempt at regulating a wind-driven alternating-current generator operating in parallel with a public-utility system. Steam at turbine and engine throttles is held at constant pressure and water to water-

wheels flows under a practically constant head or, at worst, with a slowly changing head. On the other hand, wind velocity and direction may change very suddenly; consequently, experience in governing steam turbines and waterwheels is of little account when applied to wind turbines.

The unit has unusually high unit inertia ($WR^2 \times (RPM.)^3 \div Hp$), several times that of the average hydroelectric unit, and should therefore be exceptionally stable in its operation. The only way that this characteristic can be obtained in a wind-power unit is by building in a heavy flywheel or building units of comparatively large sizes. All studies made of the problem indicated that the solution of speed regulation of wind turbines is inherently associated with large-size units. For this reason a 1000-Kw. machine is being installed instead of a small experimental unit that might not give the desired answers.

Even though the unit has very high unit inertia and should be very stable in operation it is difficult to predict what will be the effects of sudden changes in wind direction and velocity combined with abrupt load changes. Sudden gusts of wind alone might so overload the generator that it would pull out of step. The hydraulic coupling has therefore been installed to act as a slip connection and prevent sudden overloads. If this coupling is found unnecessary it may be omitted from future designs.

Wind and Water Power.

The unit will deliver power to a system supplied largely by hydroelectric plants. Wind and water power differ in several respects: Where reservoir capacity is available water can be stored during the flood period for use when flow is low. Wind cannot be stored and must be used when available like water on a river without storage. Water flow from a given area, like wind, varies widely during any one year. However, water flow also varies widely from year to year, as does the amount of power you can generate from it, while the amount of wind is fairly constant. Therefore, even though the output of a wind-power plant may vary from zero to its full capacity, the kilowatt-hours it can generate in any one year are more constant than from most water-power plants.

However, wind-power plants like this one must always operate in parallel with another source of power, such as a water-power plant with storage or a steam plant. They will probably be built in comparatively small units, of at most a few-thousand kilowatts capacity. In a national defense program this will be an advantage, because a bomb dropped on a large wind-power installation could cripple probably not more than one unit. If the same bomb fell into a 500,000-Kw. steam plant, it might shut down the whole plant for a long period.

Before a hydroelectric project can be intelligently engineered, complete knowledge of hydrology of the drainage area must be available. When this wind-power project was conceived little was known about wind characteristics at possible sites for aeroelectric plants. During the last five years much study has been given to this part of the problem by Mr. Putnam and his associates. On top of some mountains the wind is very turbulent, while on others it blows in a fairly horizontal direction. Other conditions being equal, steady horizontal flow is preferable.

In certain temperate climates ice is an important consideration. For the plant under discussion there are better sites for it from wind considerations, but the ice conditions in winter are likely to be much worse than on Grandpa's Knob; therefore it was selected to reduce risk from ice for the experimental plant.

Palmer Cosslett Putnam, who invented and worked out the general design of the wind turbine, first began to study the problem of wind power in 1935. At that time he came to the conclusion that all previous wind-power units

had been built on too small a scale to be constructed at an economical unit cost and also that their unit inertia was too low for satisfactory speed regulation. Preliminary studies made at the suggestion of Dr. Vannevar Bush, then Dean of Engineering, Massachusetts Institute of Technology and now President of Carnegie Institution, Washington, D. C., verified these conclusions. Dr. Bush suggested to T. S. Knight, vice-president, General Electric Co., that the project should be worked up in detail and Mr. Putnam then made the general design.

In 1939, Mr. Knight interested Beauchamp E. Smith, vice-president and general manager, S. Morgan Smith Co., who assumed financial responsibility for the project. Walter S. Wyman, president, New England Public Service Co., also became interested and made available the power system of the Central Vermont Public Service Corp. as the guinea pig onto which the project could be connected.

Before deciding on the final design, models were built and these were tested by Professor Elliott Reid in the wind tunnel of Stanford University, under the direction of Theodore von Kármán, director of the Guggenheim Aeronautical Laboratories, California Institute of Technology. Dr. John B. Wilbur, director of the Structural-Analysis Laboratories at Massachusetts Institute of Technology, is chief engineer of the project. Palmer Cosslett Putnam is aeroelectric consultant and project engineer.

Joseph S. Newell, Professor of Structural Aeronautics at Massachusetts Institute of Technology, was responsible for the turbine-blade structural design. With the exception of the tower all structural steel work, the pintle and main shafts and the shop assembly of the unit was done by Wellman Engineering Co. Engineering for the cooperating utility-system company is being done by the Nepsco Services, Inc., Augusta, Me.

Vision and Courage.

The technical and economic possibilities of fuel and water-power developments have been pretty well explored, but the Grandpa's Knob project is the first attempt to harness the wind for electric-power generation on a commercial scale. Those who have made this tremendous research project possible are entitled to great credit for having the vision to explore this new field of power and for having the courage to back their convictions to the limit with their time, their energy and their financial resources. Here is another example of unfettered individual initiative that has made America great and will continue to make her so as long as given free rein.

SALVAGE YOUR OWN SHIP.—Interesting details of a modern saga of the sea were disclosed in January of this year during salvage hearings in the English Courts in behalf of certain officers and crew members of the S.S. *San Demetrio.* Counsel for the defendants in this case had the happy task of declaring himself present to fully support the evidence of the claimants. The managing director of the defendant company submitted a very complimentary report on the conduct of the claimants. This article is reprinted from the pages of the May, 1941, issue of The Nautical Gazette.

Is there a right to reimbursement for salvage of one's own vessel? A legal case arising out of the heroic action of the S.S. *Jervis Bay* which, as everybody will recall, protected last year successfully a British convoy of 38 ships against a German raider, was tried before the British courts. The question was whether the owner of one of the convoyed vessels, the S.S. *San Demetrio,* was entitled to salvage with respect to his own vessel. The judg-

ment of Mr. Justice Langton gives a very vivid picture of the case, which is quoted, not only on account of its legal and factual background, but particularly in view of the tribute which the English court paid in the judgment to an American seaman.

Mr. Justice Langton in giving judgment said:

"It is now part of English history that on November 5, 1940, the convoy in charge of the British auxiliary cruiser *Jervis Bay* was attacked by a heavily armored and powerful German warship in the Atlantic. With magnificent courage and self-sacrifice the unarmored *Jervis Bay* immediately turned towards the raider, thus drawing her fire and giving her convoy the best possible chance of escape.

The plaintiffs in the present case have testified here not only to the single-minded devotion to duty exhibited by the *Jervis Bay,* but, also, to the practical value of her action. In the gathering dusk every moment that could be gained was immensely precious to the scattering convoy, and it is not in doubt that the delay so dearly purchased was also richly rewarded. The *San Demetrio* had been one of the first line of traders in the convoy, and, when the raider was able to turn her attention away from the *Jervis Bay,* was one of the first to feel the weight of her attack. She was hit by the third salvo directed at her and holed on the port bow just above the water-line. The *San Demetrio,* which is a tanker of 8000 tons gross, fitted with Diesel engines, was fully loaded at the time with a cargo of 11,200 tons of petrol. She was obviously an easy prey to the raider, who had already got her range, in addition to possessing a great superiority in speed. Accordingly, her master, Captain Waite, very properly gave the order to abandon ship, and the ship's company of 42 hands all told left the vessel in three lifeboats. One of these, the port lifeboat, contained 15 of the present plaintiffs and an engine hand, John Boyle, who died some days later. This lifeboat was commanded by Second Officer Hawkins. Just after the boats had got away, the *San Demetrio* was again struck by a heavy salvo and burst into flames. As the night wore on the weather rapidly worsened, and by dawn on November 6 a gale was running, and the boats had lost touch with one another.

For some hours the boat containing the plaintiffs was forced to lie to a sea anchor, and her crew were hard put to it to keep her head to sea with their oars. About noon a vessel, believed to be a Swedish ship, which gallantly returned to rescue the survivors of the *Jervis Bay,* was sighted, but the plaintiffs did not succeed in attracting her attention. In the afternoon, however, a tanker was seen to leeward. The boat was pulled towards her and the tanker proved to be the *San Demetrio,* still on fire amidships, and aft, and surrounded by floating petrol. In view of the imminent danger of explosion, Mr. Hawkins vetoed any close approach that night, but at dawn on November 7, finding the vessel still afloat, some five miles off in more moderate weather, he decided to hoist sail and make for the *San Demetrio.* The boat's crew experienced the greatest difficulty in boarding the burning derelict, and in spite of plucky efforts, in which the American seaman, Oswald Preston, specially distinguished himself, it was found impossible to hoist up the lifeboat. In the sea which was running there was no alternative but to allow the boat to go adrift. The plaintiffs thus found themselves in mid-Atlantic in a burning vessel, with a dinghy as their only resource if they could not succeed in extinguishing the fires and getting the *San Demetrio* under way. In spite of the fact that they had spent a day and two nights in an open boat in mid-Atlantic, in bad November weather, and that two of them were badly injured, they immediately set about the tasks in front of them with illimitable courage and resource.

The condition of the ship was pitiable in the extreme. The collision bulkhead was badly damaged by shell fire; there was a hole in her bows close to the water-line and the vessel was down by the head. The bridge and all

its surrounding structure was completely destroyed. The ship was without compasses, binnacle, charts, sextant, or any aid from instruments of navigation. There was no wireless, and the steam steering gear was gone. The hand steering gear aft was practically destroyed, but proved susceptible of repair, though only four spokes of the wheel were left, and as the rods had to be taken through distorted plates the gear could only be worked stiffly and with considerable caution. In addition, there was constant danger of explosion from the spewing of oil through the holes pierced by shell fragments in the deck. Until these could be plugged no galley fire could be lighted, and the plaintiffs had to subsist for the main part upon a diet of raw vegetables, save for certain culinary ingenuities of Mr. Pollard, the chief engineer."

* * *

"It is not necessary for me to attempt to set out in detail the individual achievements of each of the plaintiffs, but I cannot refrain from quoting two passages from a report drawn up by the manager of the defendant company, which depict better than I can hope to do the work of two of these men.

* * *

"One of the men on board who was not a Britisher, an American able seaman, Oswald Preston, is reported on in these terms:

" 'This man signed on in Halifax as a Nova Scotian, but is actually an American who wanted to get to U. K. to join one of the fighting forces. From the date of joining until the moment the raider was sighted he showed a marked disinclination for work, but from then on he came out in his true colors and was simply magnificent. He kept up the spirits of all in the boat with merry quip and jest and did the work of two men. He threw in his weight with those who were in favor of boarding the vessel again on the second day after leaving her and was one of the first two on board. It was he who went over the side again with a line round him and endeavored to secure the lifeboat, which was full of water. He was one of the first to get on the main deck and plug holes in the decks in way of gasoline cargo tanks to prevent the spirit spewing out every time vessel rolled. Decks were awash when this work was carried out and the shot hole pluggers were constantly under water and had to be secured by lifelines. No finer tribute can be paid to this man than the unanimous request made by all on board that the Red Ensign which the vessel was flying all through the action and until she arrived in port should be presented to their American shipmate.'

I dwell with pleasure upon this incident, for I am sure that seaman Oswald Preston is of a quality to treasure this bit of bunting, the only flag which remained on board the *San Demetrio*. He will remember, as we in England will remember, that it was flying at the masthead of the *San Demetrio* when she first met her troubles, that it remained flying through the worst of her misfortunes, and that it still flew when at long last, on November 16, shell-torn and fire-scarred, but undefeated, she came steaming up the sheltered waters of the Clyde.

This class of salvage is, necessarily, of a rare character. It is not often that seamen are called upon to salve their own vessel. The law on the matter was laid down with great strictness as long ago as the time of Dr. Lushington, and the law is not in doubt. Four requisites laid down by Dr. Lushington have to be fulfilled before seamen can be allowed to claim salvage in respect of their own vessel. The chief of these requisites is that the ship was properly abandoned under the orders of her master. In the present case there has never been the slightest shadow of doubt that all four requisites have been adequately fulfilled. The abandonment of this ship was not only a wise and proper act, it was the only possible course that could have been

adopted by her master. Moreover, it is clear that at the time when the vessel was abandoned no one had the slightest hope of returning to her; in fact, no one had the smallest expectation of ever seeing her again."

MELTING BY ELECTRON BOMBARDMENT.—Usually, high melting temperature alloys are made in a high-frequency induction furnace or by powder metallurgy. The April, 1941, issue of the Metallurgist, a supplement to the 25 April, 1941, issue of The Engineer (London), printed this article on another method which can be used when preparing small quantities of alloys from metals with very high melting points.

There are technical difficulties in preparing alloys from metals of very high melting point, though the successful commercial applications of many high melting point metals and alloys makes their metallographic study a matter of importance both theoretically and practically. Many of the commercial alloys are made by the methods of powder metallurgy, but production in the molten condition is likely to give a more strictly homogeneous material and is preferable for alloys intended for physical studies. To obtain the uncontaminated alloy the melting must be carried out in a vacuum or inert atmosphere in a refractory crucible which does not react with the melt.

As an alternative to high-frequency induction heating, when the amount of material is very small, R. Hultgren and M. H. Pakkala have constructed an electron bombardment furnace for high temperature melting.* In principle electron bombardment is very simple. A hot filament is placed near a positively charged crucible in a vacuum. Electrons from the filament are accelerated by the voltage drop and strike the crucible. The kinetic energy of the electrons is given up in the form of heat. Thus all the power goes directly to the crucible, which may be made as small as desired.

The temperature obtainable is limited only by the refractoriness of the crucible employed. The crucibles used by Hultgren and Pakkala were made of the pure oxides of zirconium or aluminium without a binder, and stood inside a tantalum cup (purchased from the Fansteel Metallurgical Corporation, North Chicago, Illinois), which was spot welded to a tungsten rod connected to a high potential. The crucible containing the sample was placed within the cup and the cup (not the contents of the crucible) was bombarded so that the heat was transferred through the crucible to the charge by thermal conduction and radiation.

The main difficulty encountered in operating the equipment was gas discharge of electricity caused by the gas given off by the sample and its surroundings during heating. This makes it necessary to heat slowly so that the pump may have time to remove the gas as fast as it is given off. Gas discharge forms an arc if the rate of heating is too rapid. It was found best to begin with a low voltage (perhaps 250 volts) while considerable gas is expelled with the first heat. Then voltage and milliamperage may be slowly increased while the vacuum gage is watched for signs of too rapid an increase of pressure. After a long period of disuse or after a new metal is added the first melt may require an hour or more, but conditions improve with use and the time per melt becomes considerably less. The time required will also depend on the amount of gas in the sample.

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* "Preparation of High-melting Alloys with the Aid of Electron Bombardment," by R. Hultgren and M. H. Pakkala, Journal of Applied Physics, October, 1940, Vol. II, page 643.

If the sample has a high vapor pressure at its melting point it will be difficult or impossible to melt it. Thus great difficulty was experienced with manganese and with palladium and a large part of the sample was lost before melting was accomplished. On the other hand, no difficulty was experienced with cobalt (m.p. 1480 degrees C.), iron (m.p. 1535 degrees C.), zirconium (m.p. 1700 degrees C.), vanadium (m.p. 1715 degrees C.), platinum (m.p. 1755 degrees C.), titanium (m.p. 1800 degrees C.), and iridium (m.p. about 2400 degrees C.). There were signs that vanadium, titanium, zirconium, and iridium had reacted with the zirconia crucibles. The use of thoria crucibles would be possible and might possibly avoid this trouble.

The power used for melting is rather small; thus platinum was melted with 150 watts and the tantalum cup itself melted (m.p. 2850 degrees C.) with 700 watts. The rated capacity of the electrical equipment was 1250 watts (0.5 ampere at 2500 volts). Chromium (m.p. 1810 degrees C.) and columbium (m.p. 1950 degrees C.) were not melted even at very high wattages, possibly because of the formation of strong protective coatings of oxide round the grains. Ruthenium also failed to melt, though the temperature was raised so high that the tantalum cup melted.

Alloys of platinum with tungsten, molybdenum and cobalt are being successfully prepared, by the method described, for X-ray investigation of the crystal structure. The preparation of other groups of alloys could be undertaken if the tendency of some metals to react with the available refractory crucibles could be overcome.

METAL SPRAYING WITH AN ELECTRIC-ARC GUN.—This brief note describing the Schoop metal-spraying process is copied from the June, 1941, issue of Machinery. It should be of particular interest to those engaged in repair work.

A new method of metal spraying has been developed by Dr. M. U. Schoop of Switzerland, originator of the metal-spraying process, which makes use of a spray gun utilizing an electric arc to melt the metal to be deposited. Hitherto, the metal-spraying process has depended upon a gas flame of some kind to melt the metal before being deposited on the base surface.

The new Schoop process consists essentially of short-circuiting two conducting wires which pass through the spray gun, atomizing each drop of metal melted by the resulting arc, and projecting the atomized metal by means of a compressed air blast on the surface to be metallized. A small luminous arc is formed at the breaking point, insuring the continued melting of the wires, which are constantly being fed forward by means of a turbine. Although the compressed air blast directed through the arc may be fed into the gun at pressures ranging from 60 to 120 pounds per square inch, the arc is reported to be entirely stable.

The new process of metal spraying is claimed to be highly economical and efficient. It is said that about twenty-two pounds of carbon steel or stainless steel wire can be sprayed during each hour of operation. In many cases, the pre-treatment of the surfaces by sand-blasting, heretofore required, can be dispensed with because of the increased strength of bond secured by this method of deposition. Thus, if a glass plate is electro-metallized with aluminum or steel by this process, and an attempt is made to remove the deposited metal, a layer or "skin" of glass will also be torn off. It appears that, owing to the electric arc, the temperature of the sprayed metal particles is so high that they melt into the surface against which they are propelled, rather than becoming merely a surface layer.

In combination with hard alloy wire, the electric metallizing process has been found applicable to the production of low-cost die-molds for plastics. The basis of these molds is plaster-of-Paris, wood, or other material which can be readily shaped to the required form, and this basic section is then electro-sprayed with a suitable alloy.

The normal air pressure of the process is increased at certain stages during the operation by as much as 100 per cent; and the compressed air may be mixed with nitrogen. Very hard metal sections are thus obtained.

There appear to be many possibilities for the application of this process in actual production work, as well as in various repair operations. The utilizing of different combinations of metals and also of various metal layers to secure a particular structure or mechanical property are possible avenues of experimentation.

WEIGHT-SAVING BY WELDING.—British marine publications have recently shown considerable interest in the extensive use of welding in the American shipbuilding industry, particularly by the new shipbuilding yards born of the present emergency. This note is reprinted from the March, 1941 issue of British Motor Ship.

Numerous statements have been published relating to the saving in the weight of a welded steel ship's hull in comparison with a similar riveted vessel. They have, however, rarely applied to precisely similar vessels, so that the following figures concerning riveted and welded ships of the C3 class built in the United States, of which between 20 to 30 are being built, will be of interest.

These ships, of the cargo type, are 492 feet overall, or 465 feet B.P., with a beam of 69 feet 6 inches and a draught of 27 feet 2 inches. Their deadweight capacity is 11,900 tons, including fuel, stores and water, and their gross register is in the neighborhood of 9200 tons. They are equipped with machinery of 8500 SHP.

The steel weight in the hull with the riveted ship is 3600 tons, whereas the all-welded hull weighs 3250 tons, or a saving of 350 tons, which is equivalent to 10 per cent. The detailed figures for weight, output and machinery are as under:—

Steel weight	3250 tons
Wood and outfit	1100 tons
Machinery	623 tons
Total	4973 tons

MATERIALS FOR ELECTRICAL CONTACTS.—This paper is reprinted from the April 11, 1941, issue of Engineering (London).

A paper on "Materials for Electrical Contacts," by Dr. J. C. Chaston, has been released for discussion by the Institution of Electrical Engineers. The importance of the subject is shown by the fact that contact materials of one kind or another are to be found in every electrical installation and in almost all electrical instruments. An examination was, therefore, made of the principal types of failure which can occur in light-duty and medium-duty contacts of the make and break form of construction, operating in air.

Light-duty contacts, that is, those carrying currents up to about 0.4 ampere at any voltage, can generally be broken without producing electrical wear. The sole requirement is that the contact resistance shall remain at a very

low and steady value. The factors which may affect contact resistance are design, *i.e.*, area, shape, pressure and surface finish; formation of tarnish films as the result of atmospheric corrosion; and formation of high resistance films by dust, adsorbed gases or grease. It has been well established that, when two metal surfaces are pressed together, contact takes place only at relatively few spots. For this reason the contact resistance of flat surfaces under any given pressure is largely independent of the total area. Furthermore, except under very small pressures, the surface finish has very little influence on the contact resistance. If the pressure between the surfaces is increased, the area of contact becomes larger and the contact resistance is reduced. The most obvious practical conclusion from these considerations is that it is desirable to use as high a contact pressure as possible. If a total contact load of about 1 pound is employed, there is little to choose between any of the types of construction so far as the resistance of clean contacts is concerned. When lower pressures must be employed, especially if it is not possible to seal the contacts in a dust-proof enclosure, it is advisable to use a contact so shaped that a high local pressure is produced and, moreover, is maintained in service.

Of the most commonly used types of contact, those with flat surfaces are, of course, the cheapest and probably the most widely used. The difficulty is that it is obviously impossible to ensure that there is perfect alignment of the contact surfaces. Furthermore, dust is more liable to be entrapped in contacts of this type than in those of any other construction. For this reason, they are not recommended when only light contact pressures are available. A domed surface in combination with a flat surface or two domed surfaces are much more suitable in these circumstances, since the contact pressure is then localized and less opportunity is offered for dust to lodge and prevent good contact. A domed contact operating against a flat is probably the most generally useful and is more easily aligned than two domed surfaces. The diameter of the contacts used is governed mainly by mechanical considerations. If the contacts are too small, it may be difficult to assemble the relay or other equipment in which they are fitted so that they are truly in line. In order to economize in contact metal and yet permit wide tolerances in fitting, the construction known as the crossed-wire, consisting of a flat plate with a small dome in the center for one contact and a flat plate with a smaller flat plate in the center for the other has been used. This ensures that the contact pressure is localized; moreover, such a design does not entrap dust, has all the advantages of a domed contact, and yet allows considerable reductions in the weight of contact metal. The objection to it is the difficulty of fitting the wires. It is more expensive to solder the wires in position than to affix a rivet, unless the quantities involved are large enough to warrant the use of special fixtures and equipment.

The tarnish films, which form on the surface of base metals as a result of atmospheric corrosion, are seldom simple oxides. Sulphides, sulphates, carbonates and oxides may all be formed in various combinations and degrees of hydration. Copper, brass, bronze and all other base metals and alloys cannot be regarded as suitable materials for contacts in applications where a really low contact resistance is required. Even at room temperature, a sufficiently thick film forms on copper in a few hours to increase the contact resistance many times, although excessive oxidation can be retarded by a thin layer of vaseline. At higher temperatures, copper tarnishes very rapidly, and the resistance of copper-to-copper contacts maintained at 85 degrees C. may reach 1500 microhms in 24 hours. For light-duty contacts, therefore, it is customary to employ materials which are more resistant to oxidation and tarnishing. In the accompanying table, a number of the more commonly used materials are listed according to their resistance to tarnishing, and some details are given of their physical characteristics.

Common Contact Materials Classified according to their Resistance to Tarnishing.

Material.	Approximate Hardness of Annealed Materials: Vickers Pyramid Hardness Numbers.	Remarks.
Group A.—Completely Resistant to Tarnishing.		
Pure platinum ..	39	
Commerical platinum	68	
Rhodium (electroplated).	—	Tarnishes above about 900 deg. C.
Iridium-platinum—		
10 per cent. Ir ..	110	
20 per cent. Ir ..	200	
25 per cent. Ir ..	250	
30 per cent. Ir ..	290	
Ruthenium-platinum		
5 per cent. Ru ..	130	
Platinum-gold-silver		
7 : 67 : 26	70	
Gold-silver—		
70 per cent. Au ..	72	
Pure iridium	420	Tarnishes above about 900 deg. C.
Pure gold	23	
Group B.—Resistant to Sulphide Tarnishing: Liable to Oxidation at Temperatures above about 350 deg. C.		
Palladium	63	
Copper-palladium—		
40 per cent. Cu ..	130	
Silver-palladium—		
50 per cent. Pd ..	84	
Ruthenium-platinum		
10 per cent. Ru ..	195	
14 per cent. Ru ..	245	
Group C.—Resistant to Oxidation: Liable to Sulphide Tarnishing.		
Fine silver	23	Probably best all-round contact material.
Gold-silver—		
5 per cent. Au ..	27	Little, if any, more resistant to tarnishing than fine silver.
10 per cent. Au ..	28	
30 per cent. Au ..	37	
Palladium-silver—		
5 per cent. Pd ..	33	
10 per cent. Pd ..	40	
20 per cent. Pd ..	55	
Copper-silver—		
7½ per cent. Cu ..	50	Slightly more liable to tarnishing than fine silver.
10 per cent. Cu ..	75	
20 per cent. Cu ..	108	
Group D.—Liable to Oxidation, Sulphide Tarnishing, or both.		
Tungsten	290	
Silver-tungsten ..	210	
Silver-tungsten carbide	100	
Silver-molybdenum ..	175	
Copper-tungsten ..	137-290	
Silver-nickel	—	
Silver-graphite ..	—	

When complete resistance to tarnishing and oxidation in all conditions is required, the choice is limited to platinum and gold among the pure metals. Two other noble metals, rhodium and iridium, are very nearly tarnish-resistant, developing slight oxide films only when heated to between 900 degrees C. and about 1400 degrees C. Pure platinum itself is somewhat soft and it is usual, therefore, to employ "commercial platinum." This contains 0.5 per cent of another noble metal which has been purposely added as a hardener. Commercial platinum is one of the most widely used contact materials in instruments and relays carrying small currents, and experience has proved its reliability for this purpose. It is completely immune from oxidation or tarnishing at any temperature or in any environment which is likely to be encountered, and in this respect can be regarded as a standard to which all other contact materials should be referred. It is readily worked and may be used in the form of headed rivets or machined contacts of any form. Alternatively, discs may be welded to blades or other supports.

Pure rhodium is even more inert chemically to most reagents than platinum being unattacked by *aqua regia*, but it may tarnish if heated in air to about 900 degrees to 1200 degrees C. Its high cost, and the difficulty of producing it in the form of strip or wire, make it unsuitable for use in the massive form, but the recent development of bright rhodium plating has made available a contact surface which is as tarnish-resistant in almost all contact applications as pure platinum. Electro-deposited rhodium is extremely hard and if applied properly does not chip or flake. For protecting jewelry from tarnish, deposits of a thickness of about 0.000007 inch to 0.00002 inch are commonly applied, but thicker deposits, up to about 0.00025 inch thick, are often more satisfactory for contact purposes. Pure gold is extremely soft and is not generally used for contacts. Pure iridium is expensive and difficult to obtain in a ductile form, but small discs of iridium sheet have occasionally been used for contacts. Among the tarnish-resistant alloys of these noble metals, the most commonly used are the iridium-platinum alloys containing up to 30 per cent iridium. The effect of iridium additions is to harden the platinum very greatly and these alloys are chosen when the contacts are required to withstand severe battering in service. As an alternative, some attention has recently been given to the ruthenium-platinum alloys, containing up to 5 per cent ruthenium. In order to reduce the cost, the possibility of alloying cheaper base metals with these noble metals has, of course, received attention. Some years ago, it was suggested that, provided at least 50 per cent atomic of noble metal was present, the atmospheric tarnish-resisting characteristics of the metals would be retained. Although exceptions exist, this provides a good working rule. The number of alloys which fulfill this requirement and at the same time possess suitable mechanical and working characteristics is, however, limited and, in practice, the only two which have been used to any extent are the platinum-gold-silver and gold-silver alloys. The platinum-gold-silver alloy, containing about 7 per cent of platinum, 67 per cent of gold, and 26 per cent of silver, has had a rather wide popularity, and both this and the 70 per cent gold-silver alloy can be regarded as entirely non-tarnishing. Some of the more complex 15-carat gold alloys have been used for contact brushes and generally give good service.

From the account given previously it follows that if it is possible to ensure that the limiting current is not exceeded most of the troubles which result from material transfer can be avoided entirely. The value of the limiting current for material transfer when no spark-quench condenser is fitted appears, as a first approximation, to be influenced mainly by the voltage at the instant of break and by the nature of the contact material. No detailed study of the effect of voltage on the limiting current for material transfer has been made, but its effect on the current at which arc discharge starts has been studied and the results may be regarded as a guide to the general

effect of voltage variations. It appears that when the voltage at break is low, appreciable currents may be broken without setting up an arc discharge, but the current that can be broken without arcing falls as the voltage rises and attains a limiting value when the potential reaches about 50 volts. Increasing the voltage to higher values has little, if any, effect on the limiting current. It is the voltage at break that determines the limiting current.

The value of the limiting current may vary very greatly with different contact materials and, in view of the importance of this value, it has been determined directly under various conditions for a number of metals and alloys. The method of testing has been to fit the contacts into small sturdily-built relays so that they are free from visible whipping or bouncing action at break. Oscillograph records confirm that the breaking action is rapid and clean cut, but the operation of making involves eight separate bounces, at each of which contact is made and broken during a time interval of about 0.02 second. A panel of twelve relays was used and the relays were operated from a storage battery by means of a motor-driven commutator. It was found that currents as little as 0.1 ampere below the limiting current would not give rise to more than 0.001 gramme of transfer after 400,000 operations, while currents 0.1 ampere above the limiting value would transfer 20 to 40 times as great a weight of material and would often cause interlocking. When full-wave rectified direct-current from a metal rectifier was employed, however, the limiting current was not nearly so sharply marked. It would appear most likely that the limiting conditions which define the commencement of material transfer are not dependent only on the value of the current but are influenced by the energy in the circuit.

As it is not always possible to operate contacts with currents below the limiting value, a knowledge of the amount of material transfer likely to occur in service is of great help in the choice of a contact material. In pure resistance circuits operated from a battery supply, it is found that 28 mg. of fine silver and about the same amount of commercial platinum and 30 per cent iridium-platinum are lost from the negative contact after 200,000 operations at 110 volts, with the limiting current plus 0.25 ampere. In other circuits, where electrical wear is less severe, there is a more pronounced difference between the behavior of different materials. In these conditions, the rate of transfer is proportional to the number of operations.

Choice of a contact material to interrupt heavy direct-currents at high voltage is obviously a matter of considerable difficulty, but the following general observations appear to be justified: Among the noble metals, fine silver seems to resist transfer better than any other although in some rather narrowly defined conditions (below 1 ampere with rectified direct current at 100 volts in a resistance circuit), platinum may be preferable. If the choice is broadened beyond the noble metals, tungsten becomes more valuable since it resists transfer better than any other metal or alloy. Tungsten, however, becomes coated in service with a high-resistance oxide film and the heaviest possible pressures should be used to keep contact resistance as low as possible. Tungsten, moreover, is liable to be brittle, cannot be riveted or cold worked and must be attached by copper brazing to a steel base. Among the platinum alloys the iridium-platinum, as well as the ruthenium-platinum alloys, generally resist transfer rather better than platinum. The platinum-gold-silver alloys, on the other hand, are particularly liable to transfer badly.

Among silver alloys, none of the common binary alloys, such as platinum silver, resists transfer so well as fine silver. In some conditions, however, more complex alloys appear to be markedly superior. The silver-cadmium-magnesium alloy known as Elkonium 61, has an unusually low rate of transfer with high currents under certain conditions of test. It has been suggested that in this alloy, as well as in the silver-cadmium-nickel alloy

known as Elkonium 17, transfer is hindered by cadmium and other oxides which form on the contact surfaces. These oxide films have sometimes given rise to trouble through high contact resistance, but it is claimed that a combination of Elkonium 17, as the positive contact with a palladium-silver alloy known as Elkonium 11 as the negative, possesses both low rate of transfer and freedom from high-resistance films. It has been used in some railway signal relays and low-voltage regulators in America. In medium-duty circuits, the resistance to transfer of the special composite materials such as copper-tungsten, silver-molybdenum, and silver-tungsten (including those known as Elkonites) is not so great as that of tungsten, but is generally rather better than that of silver. The silver-nickel powder metallurgy products also show considerable resistance to transfer and, in some conditions, are among the best materials which have been tested.

While the amount of transfer which occurs is largely influenced by the electrical conditions in the circuit and the nature of the contact material, it may also be affected by some more or less external conditions. The pressure with which the contacts are brought together seems to have a slight effect on the nature of the build-up. Some preliminary experiments have indicated that "spikey" build-up is liable to form more readily if the contact pressure is very light. Under heavier pressures, the build-up tends to be more dome-shaped. The size of the contact may also have a minor influence. It has been found that with fine silver the current at which build-up occurs with 110 volts rectified direct current can be raised from 0.5 ampere to 0.9 ampere by increasing the diameter of the contact head from 0.128 inch to 0.375 inch. To obtain these results it was necessary, however, to take great care to align the contact surfaces. The effect is most probably to increase the number of points at which true contact occurs. This explanation is supported by the observation that with currents above the limiting value, the loss of weight of the negative contact was independent of the diameter, although the height of build-up was less and better distributed. No improvements, however, appear to be obtained by increasing the head thickness (and thereby increasing the capacity for heat transfer) and build-up appears to be independent of the rate at which the contacts are operated or the speed at which they are separated. The nature of the atmosphere surrounding the contacts may, in many cases, influence build-up considerably. The effect of humidity, which is obviously of practical concern, does not appear to have received a great deal of study, and no reliable information concerning it seems to be available. In certain applications, as in car and aero-engine magneto contacts, the action of oil and petrol vapors needs to be considered. In these conditions, platinum and platinum-alloy contacts are liable to give considerable trouble through build-up. When these materials are used, carbon readily deposits as a closely adherent film on the contact surfaces, the platinum probably acting catalytically to promote cracking of the hydrocarbon vapors. The result is, in effect, to transform the contact surfaces from platinum, for which the limiting current is high, to carbon, for which it is very low. Build-up then commences and continues with great severity.

The values quoted for the limiting current and the rate of transfer given above apply only in the absence of spark-quenching devices. In general, it is possible to reduce arcing and material transfer in direct-current circuits very considerably and often to eliminate them entirely by shunting a condenser and series resistor across the contact gap, though the values of capitance and resistance need to be carefully adjusted to meet circuit conditions. Failure due to welding of the contacts is invariably due to a current surge on make and often gives rise to considerable trouble. The obvious steps to overcome it are:—The use of a sufficiently high value of series resistance in the spark-quench circuit, the use of as high a spring

pressure as possible to separate the contacts on break, and the use of a material having an inherently low tendency to weld. In order to obtain a relative measure of the welding characteristics of various materials the contacts were mounted in the relay previously described and were employed to interrupt a current of 1.6 amperes in a 110-volt rectified direct-current circuit. The contacts were shunted by a 2μF condenser in series with a variable resistor and adjusted so that the spring pressure separating the contacts was 90 gm. Welding of all contacts occurred when the resistance in series circuit was below 0.1 ohm. On increasing the value of resistance, however, the current surge at make was reduced and a value of resistance was finally obtained which was just sufficient to allow the relay to operate for at least 20 seconds without sticking. Wide variations in the capacitance of the condenser have no appreciable effect on the readings of this resistance, which is taken as a measure of the ease of welding of the contact material. It may be noted that commercial platinum and the iridium-platinum alloys are particularly prone to sticking. The 10 per cent gold-silver alloy is slightly superior to fine silver, and this possibly accounts for the good service which this alloy has been found to give in many applications. Similarly, platinum-gold-silver is rather less susceptible to sticking than commercial platinum. The best materials are the powder metallurgy products containing tungsten, molybdenum or nickel.

In alternating-current circuits contact troubles due to material transfer are generally absent, and welding of the contacts occurs far less frequently than in direct-current circuits. Reports of build-up in alternating current circuits have been encountered occasionally, but these may possibly be attributed to special circuit conditions, which cause the contacts to open always at the same part of the current cycle. The usual effect of alternating current is to cause a general uniform roughening of the contact faces and both contacts lose weight at a relatively low rate. For most alternating-current applications fine silver appears to be the most suitable contact material. The effect of the arc is to keep the surface relatively clean and to remove tarnish films. If, however, the utmost reliability is required, platinum contacts are probably to be preferred, since they will not tarnish if allowed to remain in the open position for prolonged periods. In choosing sliding contacts, it is generally considered that the effects of tarnish films can be neglected, since the wear between the rubbing parts will keep the contact faces clean. When very small currents are to be handled, however, the wear is not always sufficient to ensure that the lowest values of contact resistance are maintained, and it is then advisable to choose tarnish-resistant metals. The main problem is to choose two sliding-contact materials which will not wear unduly, it being noted that hardness is not always associated with the best wear resistance. In order to obtain data on the wear resistance of various combinations of metals in the absence of lubricants, a plate of one metal was pressed with a known force against a small revolving wheel of the other metal. The wheel wears a channel in the stationary plate and the length of the channel is taken as a measure of the wear. Most tests have been carried out by running at 200 RPM. for 20 minutes with a pressure of 100 gm. The length of channel varied from 0.3 mm. with a 15-carat gold alloy wheel and a rhodium plate to 1.3 mm. with a rhodium wheel and a fine silver plate.

SENSITIZED METAL TEMPLATES.—The Engineer (London) printed the following short description of the Carter template-making process in its January 17, 1941, issue.

The Carter processes were first used in England in the year 1928 for the making of templates in the manufacture of guns, automobiles and ships. With

the complete success of the method on copper, steel, brass, aluminium, plastic, wood, cloth, &c., the engineers developed the idea of not only using it for templates, but in duplicating fixtures, tools, jigs and parts for all types of metal fabrication.

In the making of a boat, aeroplane or tank the first step toward making a new model is the template development for the body group and the releasing of the lay-out of the body lines. When this is complete it becomes a master reference, permitting the next step to be made, which is that of transferring lines and contours from the loft tables on to translucent sheets used in the process. These sheets, when finished, constitute individual full-size lay-outs of the major transverse bulkheads, interspar beams or other major or minor details of the structure.

When the drawings, up to 10 feet by 12 feet, are completed the production department has already ordered the sensitized metal sheets in the necessary numbers to make up the machines. The translucent sheet with its full-scale drawing now becomes the actual photographic negative. These drawings are placed over the sensitized metal and, with a quarter of a second's exposure and three minutes' developing and fixing, the tool departments, fabrication departments and the assembly departments have any number of metal sheets that may be immediately fabricated into the finished machine.

By means of these processes the work of photoprinting from the translucent sheets on to duralumin, steel, plastic, wood, cloth or other surface is the work of a few seconds. When the photo prints are completed they are the templates that constitute the sole means of reference for the shops in the fabrication of parts and in the assembly of parts into the completed bulkhead or beam.

The old system of drawing on paper in half or quarter size and then having the metal fabricators translate the drawing to metal is completely eliminated. The new processes rapidly supply any number of full-scale completely detailed photo drawings on metal. The business of the fabrication department is rapidly and accurately to follow the lines, contour, bending and punching instructions on each sheet.

Only one objection has been raised in connection with this new process of metal fabrication. This objection comes from the template building department; the complaint is that an average draughtsman cannot accurately draw a template requiring precision within 0.001 inch. However, the fact has been demonstrated that a skillful draughtsman can draw as accurately with a sharp lead pencil on a smooth surface as a template maker can scratch with a sharp point on a metal surface.

The Metal Photo-Drafting Company, New York, is installing plants that include a complete metal inhibiting and metal sensitizing department, photo printing and developing department, and personnel to carry out the making of full-scale drawings on metal.

FRANKLIN INSTITUTE HONORS UNITED STATES NAVY.

Recognizing the achievement of the United States Navy Department in developing means for saving lives of seamen trapped in disabled submarines, The Franklin Institute awarded its Cresson Medal to the Navy this year. The presentation was made at the Institute's annual Medal Day exercises which were held on Wednesday, May 21.

Assistant Secretary of the Navy Ralph A. Bard, accepted the Medal on behalf of the research workers in the department who developed the United States Lung and Rescue Chamber.

The necessity for more effective means of rescuing men trapped in sunken submarines was brought to the forefront of public and official consciousness in December, 1927, when the S-4 sank near Provincetown, Massachusetts, with a loss of 47 men. The Bureau of Construction and Repair of the Navy Department began to study the problem from two points of view—individual escape where no outside aid is available, and collective rescue, where the location of the submerged boat is discovered in time to permit a rescue vessel with the proper equipment to remove the crew.

While the problem had been given some attention before 1929, the only devices available were a helmet, and a breathing bag in use by the Germans. For collective rescue the raising of the vessel by means of pontoons had to be relied upon. Tests were made on existing equipment, which, for a number of reasons, was found unsatisfactory.

Tests conducted by the Navy in a 60-foot deep tank at the Washington Navy Yard gave valuable information concerning the practicability of rescue devices which would be small and light enough to permit of carrying a full supply on a submarine; whose operation would be simple and possible under all conditions, and which would be effective in rescue within the physical limitations of the human body and with respect to the dangers of decompression.

The United States Lung was the answer to the problem, and with it ascents have been made from depths of 200 feet. Training in practice tanks under conditions closely resembling those of actual sinkings is now a part of the standard qualifications of candidates for submarine duty.

The approach to the solution of the collective rescue problem came through a study of "surface assistance" possibilities. Several types of rescue chambers were developed and tested, with the result that a new type was perfected. This was used in the successful rescue of all the living seamen on the ill-fated *Squalus* when she sank in 240 feet of water in May, 1939. Twenty-six men were brought to the surface in the rescue chamber, while thirty-one were lost due to the flooding of a compartment at the time of the accident.

BOOK REVIEW.

By Lt. Commander M. M. Dana, U. S. N.

THE HIGH-SPEED COMPRESSION-IGNITION ENGINE.—By C. B. DICKSEE, M. I. Mech. E., M. I. A. E., F. Inst. Pet.—Blackie & Son Limited, London and Glasgow, England, 1940. Cloth, 6x9 in., 331 pp., Illus., Tables,—$4.50.

1. This volume is a discussion of processes involved in charging the cylinder of a high-speed Diesel engine and introducing the fuel, and also of types of combustion chambers and their effect on the power, pressures and temperatures developed.

2. The author contends that the modern high-speed engine is not a direct offspring of the older, heavier slow-speed engine, but is an entirely new development which required a tremendous expenditure of energy and money. He points out, however, that lessons learned by working with the high-speed engine are directly applicable to the slower engine and if properly applied should result in improved performance of these latter type engines.

3. The book opens with a brief review of the thermodynamic laws of gasses and the idealized cycles of operation applicable to internal combustion engines, namely, constant volume (Otto) cycle, constant pressure (Diesel) cycle and the mixed cycle, which is a truer criterion of what happens in the high-speed engine than either of them. The chemistry of combustion is next discussed and the author introduces the novel concept of appraising the operation of the engine on the basis of per cent of air consumed, rather than on the more conventional air-fuel ratio. He thereby emphasizes the fact that an engine will produce more horsepower if it can be made to consume a large portion of the air within its cylinders.

4. The losses and limitations of the practical engine are analyzed next and the reasons why theoretical efficiency is not attained are given. The relative importance of each type of loss, such as

exhaust losses, losses to cooling water, effect of the time required for completion of combustion, dissociation, etc., are discussed.

5. A chapter is devoted to air charging and the effects of valve timing, intake restrictions including valve marks, heat transmission from cylinder walls, and effective compression ratios are examined. The process of combustion is investigated by breaking it up into the accepted three phases, viz., delay, uncontrolled conflagration and controlled combustion. The effect of engine speed on combustion is analyzed.

6. An interesting portion of the book is that relating to air movement. The necessity for an air swirl is brought out and various means for producing it are discussed. This naturally leads to combustion chambers which the author discusses under four headings; open chamber, pre-combustion chamber, swirl chamber and air-cell chamber.

7. Existing systems of fuel injection, constant pressure, spring injection and cam operated plunger systems and the relative merits of each for various types of engines and classes of service are analyzed. Typical trouble with each injection system are discussed and the effect of variables under the control of the designer, such as open versus closed nozzle, residual pressure, speed delivery characteristics of pump and pipe length and size, are analyzed.

8. The final chapter deals with "Some Practical Results" and includes actual operating data which the author considers illustrative of modern practice. Methods for comparison of results from different engines wherein the variables are reduced to common denominators are presented. The closing paragraph on "future developments" is an interesting summary of what has already been accomplished and in what direction further accomplishments are most likely.

9. The book presents a welcome summary of recent developments and prevailing theories discussed in current literature of the last few years and should be read by every engineer engulfed by the rapid developments of the high speed Diesel engine.

SEA POWER IN THE MACHINE AGE—By BERNARD BRODIE. Published by Princeton University Press, Princeton, N. J. Price $3.75.

Sea Power in the Machine Age covers the period of development of Naval vessels which is best described by the titles of the five parts:

Part I The Steam Warship.
Part II The Iron Hulled Warship.
Part III Armor Versus Great Ordnance.
Part IV The Torpedo, The Mine and The Submarine.
Part V Aircraft in Naval War.

The author adheres rigidly to his conception of Naval inventions of real importance as those described by the above five titles. Subsidiary developments are described, but treated as strictly subsidiary to the main "invention." The treatment of these five "inventions" as to their effect on Sea Power in the sense of its broadest definition is very thorough and for its thoroughness convincing.

The discussion includes effects in the current war as late as early 1941. Conclusions reached may necessarily be changed before this war is concluded, but surely the author has overlooked very little evidence up to the very latest moment.

The book may initiate some changed trains of thought among professionals, but will teach them little. The digestion by the layman and amateur "Grand Strategist" will benefit the nation.

ASSOCIATION NOTES.

In connection with the preparation of a new Membership List issued with this JOURNAL, it has developed that some of the Civil and Associate Members of the Society have attained a Naval Reserve Status. The By-Laws provide that Reserve commissioned and warrant officers of the Navy shall be eligible for Naval Membership. In the event that any member who has the necessary status desires to be transferred to Naval Membership, his application for such transfer should be submitted.

MEMBERSHIP.

The following have joined the Society since the publication of the May, 1941, JOURNAL:

NAVAL.

Brown, James Harvey, Lieutenant, U. S. Navy.

Cameron, Donald A., 55 Hanson Place, Brooklyn, N. Y.

Colie, Runyon, Jr., Ensign, U. S. N. R., 2345 Rusk St., Beaumont, Texas.

Dawson, William A., Ensign, U. S. Navy.

Flewelling, Milton F., U. S. Maritime Commission, Washington, D. C.

Fulton, Robert B., Lieutenant, U. S. Navy.

Gallant, Alfred Edward, Jr., Ensign, U. S. Navy.

Herr, Donald Lincoln, Ensign, U. S. N. R., Bureau of Ships, Navy Dept., Washington, D. C.

Hoffberg, Howard J., Ensign, U. S. N. R., Bureau of Ships, Navy Dept. Mail, 3583 Warder St., N. W., Washington, D. C.

Lee, John Carlton, Ensign, U. S. N. R., U. S. S. *Spica*.

Levy, Richard S., Ensign, U. S. N. R., 11 Regent Circle, Brookline, Mass.

Mullally, Joseph W., Anaconda Wire & Cable Co., Room 1870 25 Broadway, New York, N. Y.

Pape, Charles G., Lieutenant, U. S. N. R., Director of Conversion & Repair, Third Naval District, 90 Church St., New York, N. Y. Mail, 7615 First Ave., North Bergen, N. J.

Platt, George F., Lieutenant, U. S. N. R., care Inspector Naval Material, 35 E. 7th St., Cincinnati, Ohio.

Schrager, Victor L., Ensign, U. S. Navy.

Spencer, Frank A., Ensign, U. S. N. R., Office of Asst. District Material Officer, 8th Naval District, Galveston, Tex.

White, Norman W., Ensign, U. S. Navy.

CIVIL.

Brush, Gorton William, Plant Engineer, Ingalls Shipbuilding Corp., Pascagoula, Miss.

Gloetzner, Alwin A., New Departure Division, General Motors Corp., 1421 Massachusetts Ave., N. W., Washington, D. C.

Jenkins, James M., Vice President and Sales Manager, Andale Co. Mail, 6341 Greene St., Germantown, Philadelphia, Pa.

McNeal, D. Raymond, President and General Manager, Andale Co. Mail, 308 Jericho Road, Abington, Pa.

Rodenbaugh, Henry N., Vice President, Day & Zimmerman, Inc., Packard Building, Philadelphia, Pa. Mail, Seven Pines, Elbow Lane, Haverford, Pa.

Scannell, T. F., Sales Manager, The Falk Corporation, Milwaukee, Wis.

Schoenfeld, David M., Combustion Engineering Company, 200 Madison Ave., New York, N. Y.

Terry, R. V., Assistant Chief Engineer, Newport News Shipbuilding and Dry Dock Co., Newport News, Va. Mail, P. O. Box 477, Hilton Village, Va.

ASSOCIATE.

Apel, Arno A., General Manager, Ventnor Boat Works, Inc., 228 N. Derby Ave., Atlantic City, N. J.

Burns, Charles Ira, Assistant Marine Engineer, Office of Superintendent Shipbuilding, New York, N. Y. Mail, 1012 Ocean Ave., Brooklyn, N. Y.

Cochrane, Herbert A., Treasurer, Electric Heater Company, Box 1082, Bridgeport, Conn.

Daley, J. R., General Manager, Shipbuilding Division, Willamette Iron and Steel Corporation, Portland, Oregon. Mail, 3050 N. W. Front Ave., Portland, Ore.

Hazen, Edwin Ray, Manager, Washington Office, Sterling Engine Company, Room 806 Evans Building, Washington, D. C.

Heller, Franklin J., Associate Marine Engineer, Design Section, Navy Yard, New York. Mail, 59-73 60th Place, Maspeth, L. I., N. Y.

Jones, William K., Chief Engineering Draftsman, Piping Group, Navy Yard, Puget Sound, Bremerton, Wash.

Kampf, Henry, Executive Assistant to President, Combustion Engineering Co., New York, N. Y. Mail, Wardman Park Hotel, Washington, D. C.

McRae, Duncan, Chief Engineer M. C. *Cucuta,* Cartagena, Colombia, S. A.

Simmons, Edward E., Jr., Research Fellow, California Institute of Technology, Pasadena, Calif.

Sneed, Harold E., Chief Engineer, DeWalt Products Corporation, Lancaster, Pa.

Ulrich, William, President, American Oil & Supply Co., 238 Wilson Ave., Newark, N. J.